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Experimental Investigation of the Melting Behavior and the Transient Heat Transfer in a Phase Change Material (PCM)

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Abstract

Phase change materials (PCMs) used in thermal energy storage (TES) system as heat storage media can play crucial role in various engineering applications. PCMs have the inherent ability to store large amount of heat by exploiting their heat of fusions, predominantly during their solid-liquid phase-change period. The present study reports an experimental investigation on the melting process in a PCM and the associated heat transfer in a transient manner. The experiments were conducted in a rectangular chamber filled with paraffin wax used as the PCM and, a cylindrical copper tube with different geometries and orientations as the heat source.

The simultaneously recorded temperatures and the images enabled to analyze the spatio-temporal behavior of the PCM during melting and provided a platform to characterize various important thermophysical parameters associated with the phase transition process. The results show a nonlinear melting rate of the PCM and a wavy and non-uniform profile of the solid-liquid interface. The results also show the development of convective motions within the liquid PCM which influenced the local heat transfer coefficient in a transient manner and is believed to have immensely influenced the nature of this distinctive and non-linear pattern of the solid-liquid interface. The convective heat transfer coefficient, \( h \) increased sharply at the early stage of melting, followed by a more gradual increment in \( h \), which then became nearly steady when the melted fraction reached approximately two-third of the total PCM volume. The continuous change in the structures of the flow inside the convective cell is believed to have significantly influenced the heat transfer coefficient. The results enabled to characterize various important thermophysical correlations in non-dimensional form associated with the PCM during the melting process to comprehend the convective heat transfer behavior and to extend the understanding on the interface dynamics. These correlations are found to have similar trends compared to other empirical correlations found in the literature.

On the other hand, most PCM suffer from the inherently common problem of low thermal conductivity, which hampers in the heat transfer rates. This study also focused on the understanding of thermal conductive enhancement in PCM using nanoparticles mixed with
Results show substantial improvement in various thermophysical parameters of the PCM with a small percentage of nanoparticles added.

Keywords

Thermal energy storage, PCM, latent heat, natural convection, convective heat transfer coefficient, solid-liquid interface, thermal convective enhancement, high speed imaging
Co-Authorship Statement

I hereby declare co-authorship in the chapters listed below:

**Chapter 2:**

Characterization of the melting behavior and the transient heat transfer in a phase change material

Authors: Mohammad Bashar, K. Siddiqui

Status: The paper will be submitted to a journal in the area of thermal sciences.

Experimental work and data analysis were performed by Mohammad Bashar. The manuscript was written by Mohammad Bashar and reviewed by K. Siddiqui.

**Chapter 3:**

Natural convection melting process in a PCM based thermal energy storage with a U-shaped heat source

Authors: Mohammad Bashar, K. Siddiqui

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**Chapter 4:**

Investigation on the heat transfer enhancement in a phase change material using nanoparticles

Authors: Mohammad Bashar, K. Siddiqui

Status: The paper will be submitted to a journal in the area of thermal sciences.

Experimental work and data analysis were performed by Mohammad Bashar. The manuscript was written by Mohammad Bashar and reviewed by K. Siddiqui.
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Nomenclature

Notations / List of symbols:

\( A \) \quad \text{Surface area of heat source, m}^2

\( C_p \) \quad \text{Specific heat capacity, kJ/kg/K}

\( D \) \quad \text{Diameter of the heat source tube, m}

\( Gr \) \quad \text{Grashof number}

\( g \) \quad \text{Acceleration due to gravity, m/s}^2

\( H \) \quad \text{Height of the chamber, m}

\( \Delta h_m \) \quad \text{Latent heat of fusion, kJ/kg}

\( h \) \quad \text{Convective heat transfer coefficient, W/m}^2/K

\( k \) \quad \text{Thermal conductivity, W/m/K}

\( l \) \quad \text{Characteristic length, m}

\( L \) \quad \text{Length of the chamber, m}

\( \dot{m} \) \quad \text{Mass flow rate, kg/s}

\( Nu \) \quad \text{Nussult number}

\( Pr \) \quad \text{Prandtl number}

\( q \) \quad \text{Thermal energy, W}

\( q'' \) \quad \text{Heat flux, W/m}^2

\( Ra \) \quad \text{Rayleigh number}

\( Re \) \quad \text{Reynolds number,}

\( t \) \quad \text{Time, s}

\( T \) \quad \text{Temperature, K}

\( T_{amb} \) \quad \text{Ambient temperature, K}
\[ T_m \] Melting temperature, K
\[ T_{surf} \] Surface temperature, K
\[ x/L \] Dimensionless length (horizontal)
\[ y \] Characteristic height, m
\[ y/H \] Dimensionless height (vertical)

**Subscripts**

- \( bf \) Base fluid
- \( c \) Charge
- \( d \) Discharge
- \( f \) Fluid
- \( i \) Inlet
- \( l \) Liquid
- \( local \) Local value
- \( m \) Mean/melt
- \( m_f \) Fraction melted
- \( nf \) Nano fluid
- \( o \) Outlet
- \( w \) Wall (tube)
- \( s \) Steady state
1  initial value
2  Final value

Greek symbols

$\beta$  Volumetric thermal expansion coefficient, $1/K$
$v$  Kinematic viscosity, $m^2/s$
$\alpha$  Thermal diffusivity, $m^2/s$
$\delta$  Boundary layer thickness, m
$\mu$  Dynamic viscosity, $kg/m.s$ or $N.s/m^2$
$\rho$  Density of fluid, $kg/m^3$

Abbreviations

DI  Deionized Water
EG  Ethylene-Glycol
GHG  Greenhouse gases
HTF  Heat Transfer Fluid
Preface

The man who removes a mountain begins by carrying away small stones.

– Chinese Proverb

It does not matter how slowly you go as long as you do not stop.

- Confucius.

To those who are relentless in creating a cleaner environment

for the future generations
Chapter 1

1 General Introduction

1.1 Introduction

The escalating global warming as a result of the detrimental effects of the greenhouse gas (GHG) emissions and major concerns over the rapidly decreasing conventional fossil fuel reserve are the foremost driving forces behind efforts on how more effectively and efficiently various energy resources can be integrated in the current energy pattern [1]. Furthermore, with a substantial increase in the energy demand due to industrial and commercial developments, population growth and improved lifestyle, it is becoming extremely imperative to efficiently manage energy consumption and minimize energy waste. Considerable work has been continuing for over three decades to restructure the energy mix to utilize renewable energy resources with the conventional resources to reduce dependency on the latter [2]. Some of the prolific renewable energy sources, for example solar, wind, geothermal, bio-mass and marine/ocean energies are considered not only the key contributors in the sustainable energy perception, but are the front-liners for providing clean energy [3]. But, despite having tremendous potential, these renewable energies have failed until present to provide a very effective alternative platform to conventional fuels because of their low conversion factors and, certain geo-physical constraints beyond human control. Few significant examples of the latter may include 1) unpredictable availability, 2) mismatch between supply and demand, 3) large discrepancies in the energy intensities due to seasonal variations, and 4) disparities due to the geographical locations.

Energy in any form is precious and delicate. But, converting them in a usable form is undoubtedly extremely important and valuable. Thus, from every perspective, its production and use in an efficient way is one of the ultimate goals behind any research and development work. Increasing demand for energy is an obvious reality whereas the conventional resources are limited and exhausting at an alarming rate. Besides huge efforts on how more and more alternative energy resources can be mixed with the present use
pattern, the rate of induction of renewable energies are significantly lagging and as figure 1.1 shows, it has been predicted that the fossil fuels (coal, oil and gas) will continue to be the primary provider of over 80% of the total energy supply globally [4]. The projected trend does not even show sign of any appreciable enhancement in renewable energy’s share in the near future. A major change is not expected unless there is a remarkable improvement in the fundamental understanding on the energy extraction and conversion process, progress in the current clean technologies to make them more efficient and cost effective, development in material science and substantial changes in the performance and energy utilization patterns.

![Figure 1.1: Primary energy consumption by fuel type between 1980-2040 (quadrillion Btu) [4]](image)

Energy production and consumption in an efficient way is tied to all urban and industrial activities including all future developments since they are the largest stake holders among energy users. Since none of the renewables can be employed practically as a ‘stand-alone’ energy source, as per the context of the present technology, the best option is to utilize these resources in parallel with the conventional systems followed by continued efforts to increase their (renewable energies) share in the energy mix. Hence, the pressure may gradually be relieved from the fossil fuels and undoubtedly, this transfer to less-polluting
energy sources will bring substantial benefits to the environment. But for the overall economy to get this benefit, the challenges to the current technological and physical barriers need to be analyzed and scaled.

On the other hand, wasted heat energy from industrial, commercial and residential sectors accounts for a substantial amount of energy loss when compared to the quantity produced. Ettouney et al. [5] mentioned that over one-half of the estimated input energy in the industrial sector is discarded as waste heat with flue gases or through other forms of low-grade energy. Although the amount of thermal energy loss may vary with the size or type of equipment, but in true sense, for any thermal application heat loss is unavoidable due to the low efficiency of the thermodynamic cycle or process involved. Hence, this wasted heat, or alternately, the rejected heat is actually a part of any thermal system’s inherent design. Considering the scale at which various thermal systems are utilized in various sectors, even if a small percentage of wasted thermal energies in these systems could be recovered and reused, it would be a massive benefit, both environmentally and economically [6].

When comparing various fields that utilize thermal energies, the statistics shows that the energy required for space heating and cooling purposes account for a major portion of the total energy use, especially in countries with extreme climatic condition (both cold and warm countries); this proportion can reach as high as 40% [7, 8]. In another statistic, it is reported that the energy consumption in the residential sector accounted for more than 70% of the total amount of energy consumed by buildings worldwide [9]. A projected trend shows that by 2020, the use of energy in buildings will account for more than 35% of the national energy consumption in China, of which, 65% will be consumed by HVAC (heating, ventilation and air-conditioning) systems [10-12].

In this perspective, it is important to mention that some of these renewable energies can provide thermal energy directly (for example solar, geothermal, bio-mass etc.) and some indirectly (wind, ocean etc.). From these discussions, it is imperative that creating ways for enhanced penetration of various alternative energy resources are mandatory tasks for the
present generation to secure a safe environment for the future. This was one of the top agendas in the 2015 Paris protocol too [13].

But, in spite of all these hard works, it still remained a very complex subject and a great deal of challenge for the scientists and engineers in making best use of such prospective renewable energies and reutilize the wasted heat in a productive way. Since some of the geo-physical constraints of potential renewable energy sources, as mentioned earlier, are beyond human grasp, it is thus necessary to consider how these issues can be handled. The most prolific issues of unpredictable availability, mismatch in demand and supply, seasonal variations and geographical location factors and finally, the wasted heat capturing can be solved simply by introducing, ‘Energy Storage’. Although energy can be stored in various forms, a thermal energy storage, based on application areas, carries a significant share in the overall energy storage categories, and is the primary player to recover and reuse waste heat.

1.2 Thermal energy storage (TES)

Heat and thermal energy are often used as synonyms but based on strict thermodynamic definition, they are not the same. Thermal energy describes the thermal content of a substance or a system, while, heat is the content of thermal energy that is transferred from one material or system to another as a consequence of the temperature difference between the two. Thermal energy can also be generated, at the expense of converting energy from one form to another. In general, temperature is an indicator which determines the ability for a physical system (i.e. certain amount of mass of a substance) to transmit thermal energy to another physical system. Presence of heat can only be realized when there is a difference in temperature and the energy crosses the system boundary. Thus, the heat transfer deals with the determination of the rates of thermal energy transfer by taking into account the variations in temperature.

TES system is in practice for centuries. It is essentially a means by which thermal energy is ‘hold’ for a certain period of time. Storing thermal energy (in the form of heat or cold) in an appropriate form and in a suitable media when it is surplus, and extracting the same
at a later time when needed is the basic principle of a TES. TES is proved to be an effective and practical solutions to address the issues of intermittency and can play a vital role in efficient energy management system [14].

The main purpose of TES is to overcome the mismatch between the energy generation and its utilization [15-17]. TES is gaining enormous importance and is becoming a noteworthy topic due to its growing importance in various areas of engineering and general applications. Recently, TES has attracted attention in providing heat or cold in remote places using mobile heat storage units which can offset the geographical location issues [18]. TES has the enormous potential in conserving energy which in turn can minimize the carbon footprint on the environment.

The design of a TES depends on the purpose of its use and the intended duration of storage period [19]. TES is typically a chamber where, depending on the operational criteria, appropriate material(s) is used as the medium to store the thermal energy by taking advantage of some of its thermo-physical properties. The available thermal energy is routed to the TES via a suitable medium (usually a fluid) where the heat transfer takes place and the storage material absorbs and holds the thermal energy. When required, the same phenomenon is reversed and the thermal energy is extracted from the storage material.

Figure 1.2 shows the overall function of the TES by splitting the charging, storing and discharging processes in separate schematics to clarify the functions. Among these three processes, schematic A represents the charging cycle, schematic B the storage, and schematic C is the discharge cycle. The heat charging rate is,

\[ Q_c = \dot{m}_c \cdot C_{pc} \cdot (T_{ci} - T_{co}) \]  

(1.1)

and the heat discharging rate is,

\[ Q_d = \dot{m}_d \cdot C_{pd} \cdot (T_{di} - T_{do}) \]  

(1.2)

where, \( c, d, i \) and \( o \) denotes charging, discharging, inlet and outlet respectively. The total amount of heat stored in the material (PCM for this study) from its solid state at temperature
$T_s$, passing through the complete melting stage and reaching a liquid temperature of $T_l$ is shown in equation 1.5.

![Diagram of TES processes](image)

**Fig 1.2:** A schematic of the three processes in a TES system shown separately, Charging (A), Storing (B) and discharging (C). $Q$ indicates that heat is infiltrating and the figure illustrates during the heat storing process.

### 1.3 Benefits of a thermal energy storage (TES)

The primary benefits of a TES are summarized below:

a) Efficiency: help achieve a better and efficient use of energy

b) Environmental effects: lower environment pollution, less GHG emissions

c) Sustainability: improved system performance and reliability via inclusion of more renewable energies

d) Economics: large savings following reduction in capital and operational costs

The results from a study on the potential of energy saving by using thermal energy storage in European countries [20] are summarized in Table 1.1. The study was conducted on commercial buildings and industrial sectors. The results also indicate the remarkable reduction in CO$_2$ generation by implementing TES.
Table 1.1 Potential energy savings by using TES [20]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Spain</th>
<th>Germany</th>
<th>EU-25</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load reduction</td>
<td>MW\text{\textsubscript{therm}}</td>
<td>541,266</td>
<td>480,844</td>
<td>5,854,139</td>
</tr>
<tr>
<td>Replaced thermal energy</td>
<td>GWh\text{\textsubscript{therm}}</td>
<td>826,263</td>
<td>662,291</td>
<td>9,527,227</td>
</tr>
<tr>
<td>Replaced electrical energy</td>
<td>GWh\text{\textsubscript{elect}}</td>
<td>3431</td>
<td>n/a</td>
<td>17,526</td>
</tr>
<tr>
<td>CO\textsubscript{2} emissions reduction</td>
<td>MT</td>
<td>207.67</td>
<td>165.57</td>
<td>2579.09</td>
</tr>
</tbody>
</table>

1.4 Physical methods of thermal energy/heat storage

Good heat exchange between the heat source/sink and the PCM is an essential requirement for the efficient thermal performance of a thermal storage unit. The mechanism of heat transfers in a heat exchange or an energy conversion process if very complex and especially if the process happens in a multiphase system. There are three basic and distinct modes of heat transfer used for storing thermal energy; the sensible heat, the latent heat or the thermochemical energy [21]. The sensible component accounts for the translational, rotational and vibrational motions of the atoms and molecules. The latent component is the consequences of the intermolecular forces of the material that influence the phase change between solid, liquid and gaseous states. The chemical component accounts for the energy stored in the chemical bonds between atoms.

1.4.1 Sensible heat storage

When the energy is stored or released by increasing or decreasing the temperature of the storage material respectively, it is considered as the sensible heat storage. The amount of thermal energy stored is dependent on the temperature change in the material, mass of the material, \( m \) (kg) and its heat storage capacity \( C_p \) (kJ/kg.K) and can be expressed as [22]:

\[
Q = m \int_{\tilde{T}_1}^{\tilde{T}_2} C_p \, dT
\]  

(1.3)
Where, \( T_1 \) and \( T_2 \) are the lower (initial) and upper (final) temperature levels between which the storage system is operating.

### 1.4.2 Latent heat storage

Latent heat storage system involves material’s phase transition i.e. the energy is stored or released due to the change in the enthalpy of the phase change. Due to the involvement of the phase change, the thermal energy storage operates almost isothermally at the phase change temperature of the material. The amount of thermal energy stored, \( Q \) in this case depends upon the mass, \( m \) and the latent heat of the phase change of the material and is expressed in the form [22]:

\[
Q = m \cdot \Delta h_m
\]  

(1.4)

Where \( m \) is the mass of storage material (kg), and \( \Delta h_m \) is the phase change enthalpy or alternately, the latent heat of fusion/solidification (kJ/kg).

### 1.4.3 Total amount of heat storage

When a material is heated from its solid state to complete melting and beyond (over a range of temperatures \( T_s \) to \( T_i \) that passes through the melting point \( T_m \)), the total thermal energy stored in the form of heat can be calculated as [22]:

\[
Q = m \left\{ \int_{T_s}^{T_m} C_{ps} \, dT \right\} + \Delta h_m + \left\{ \int_{T_m}^{T_l} C_{pl} \, dT \right\}
\]

(1.5)

where \( C_{ps} \) and \( C_{pl} \) represents the specific heats of the solid and liquid phases and \( T_s, T_i \) and \( T_m \) are the temperature of the solid, liquid and melting/fusion of the material respectively. During the initial stage, the PCM behave as a sensible heat storage material where its temperature increase as they absorb heat. But when the PCM reaches its melting temperature, unlike conventional sensible heat storage material, the PCM absorbs large amount of heat at nearly a constant temperature, which is in fact, the latent heat of fusion of that particular PCM. The PCM continue to absorb heat maintaining this near constant
temperature until the PCM is completely melted to the liquid phase. After complete melting, the liquid PCM temperature again starts increasing above its melting temperature like as a sensible heat storage and continue to rise depending on the input temperature. Theoretically, all phase-change processes can be employed for storage and release of energy. There are four types of changes between phases that are of significance: solid-liquid (melting and solidification), liquid-gas (evaporation and condensation), solid-gas (sublimation and re-sublimation), and solid-solid transformations. The absolute value of latent heat of phase change for a solid-solid transition is usually less than that of solid-liquid conversation. Solid-gas and liquid-gas transitions are generally not employed for energy storage, in spite of their high latent heat, only because of their (gases) occupations of large volumes, requiring a big space for storage or conversely stored at high pressure to keep smaller volume, which introduces critical and costly constraints on the storage design. Hence, a solid-liquid transition, which involves only a small change in volume and negligible pressure issues, is quite promising for phase change energy storage [23, 24]

The phase change materials (PCMs) are generally divided into two groups: organic and inorganic. Inorganic compounds show a volumetric latent TES capacity twice that of organic compounds. But the organic substances could serve as important heat storage media because of several advantages from their natural properties like, ability to melt congruently, self-nucleation, non-toxic and non-corrosiveness, environment friendly etc. Alkanes and paraffin are in the latter category [25]. Figure 1.3 shows various common materials for sensible and latent heat storage systems.
1.5 Comparison: Sensible and latent heat TES

Each system has its own advantage and disadvantage, which may vary drastically with changes in the thermophysical conditions as well as the adaptability. Thus, it is difficult to generalize the superiority of one over the other. Several explicit factors in a given application dictates the suitability and selection of a heat storage type. These factors may include type, size, location, duration, frequency, repeatability etc. of the TES’s use. In other words, choice of TES is exclusively need specific.
Although the heat transport and the design philosophy of a latent heat thermal storage system (LHTES) is much more complex than sensible heat thermal storage system (SHTES), from the technical point of view, LHTES has some major advantages over sensible type. Firstly, LHTES has the advantage of storing large amount of heat (energy storage density) in the same amount of mass of the thermal storage media by exploiting the enthalpy of phase change. Secondly, an attractive feature of a LHTES system is that, the operating temperature can be anchored in a narrow band around the phase-change temperature of the media [28], making the system very suitable for most of the applications. Another notable advantage the LHTES possess is that, even if there is a small temperature difference (ΔT) between the sink and the source, a large amount of heat can still be possible to store. Whereas, the effectiveness of a SHTES system solely depends on the temperature differences, i.e. higher the ΔT, more is the heat storage as seen in figure 1.4.

![Figure 1.4: A schematic of sensible and latent heat storage principles](image)

SHTES has the advantage of being relatively simple in design and cheap, but the energy storage density is low and there is a gliding discharging temperature that immensely affect its performance. LHTES is found in a wide range of applications, which include space heating and cooling, metal processing, thermal energy storage, temperature control in miniature electronics devices, food processing and storage, spacecraft thermal system applications etc. [29-33]. A comparison of energy storage density and typical materials used as PCM are shown in Table 1.2 and 1.3.
<table>
<thead>
<tr>
<th>Type of Storage Technology</th>
<th>Material</th>
<th>Energy stored (MJ/m³)</th>
<th>Energy stored (kJ/kg)</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensible heat</td>
<td>Granite</td>
<td>50</td>
<td>17</td>
<td>$\Delta T = 20,\text{°C}$</td>
</tr>
<tr>
<td></td>
<td>Water</td>
<td>84</td>
<td>84</td>
<td>$\Delta T = 20,\text{°C}$</td>
</tr>
<tr>
<td></td>
<td>Water</td>
<td>306</td>
<td>330</td>
<td>$T_{\text{melting}} = 0,\text{°C}$</td>
</tr>
<tr>
<td>Latent heat</td>
<td>Paraffins</td>
<td>180</td>
<td>200</td>
<td>$T_{\text{melting}} = 5\text{-}130,\text{°C}$</td>
</tr>
<tr>
<td></td>
<td>Salt hydrates</td>
<td>300</td>
<td>200</td>
<td>$T_{\text{melting}} = 5\text{-}130,\text{°C}$</td>
</tr>
<tr>
<td></td>
<td>Salt</td>
<td>600-1500</td>
<td>300-700</td>
<td>$T_{\text{melting}} = 300\text{-}800,\text{°C}$</td>
</tr>
<tr>
<td>Chemical reaction</td>
<td>$\text{H}_2$ gas oxidation</td>
<td>11</td>
<td>120,000</td>
<td>300 K, 1 bar</td>
</tr>
<tr>
<td></td>
<td>$\text{H}_2$ gas oxidation</td>
<td>2160</td>
<td>120,000</td>
<td>300 K, 200 bar</td>
</tr>
<tr>
<td></td>
<td>$\text{H}_2$ gas oxidation</td>
<td>8400</td>
<td>120,000</td>
<td>20 K, 1 bar</td>
</tr>
<tr>
<td></td>
<td>Fossil gas</td>
<td>32</td>
<td>-</td>
<td>300 K, 1 bar</td>
</tr>
<tr>
<td></td>
<td>Gasoline</td>
<td>33000</td>
<td>43200</td>
<td>-</td>
</tr>
<tr>
<td>Electrical Storage</td>
<td>Zn/Mn oxide battery</td>
<td>-</td>
<td>180</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Pb battery</td>
<td>-</td>
<td>70-180</td>
<td>-</td>
</tr>
</tbody>
</table>
### Table 1.3 Typical materials used in latent heat TES storage [16]

<table>
<thead>
<tr>
<th>Material</th>
<th>Melting temperature, °C</th>
<th>Melting enthalpy, MJ/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water-salt solutions</td>
<td>-100-0</td>
<td>200-300</td>
</tr>
<tr>
<td>Water</td>
<td>0</td>
<td>330</td>
</tr>
<tr>
<td>Clathrates</td>
<td>-50-0</td>
<td>200-300</td>
</tr>
<tr>
<td>Paraffins</td>
<td>-20-100</td>
<td>150-250</td>
</tr>
<tr>
<td>Salt hydrates</td>
<td>-20-80</td>
<td>200-600</td>
</tr>
<tr>
<td>Sugar alcohols</td>
<td>20-450</td>
<td>200-450</td>
</tr>
<tr>
<td>Nitrates</td>
<td>120-300</td>
<td>200-700</td>
</tr>
<tr>
<td>Hydroxides</td>
<td>150-400</td>
<td>500-700</td>
</tr>
<tr>
<td>Chlorides</td>
<td>350-750</td>
<td>550-800</td>
</tr>
<tr>
<td>Carbonates</td>
<td>400-800</td>
<td>600-1000</td>
</tr>
<tr>
<td>Fluorides</td>
<td>700-900</td>
<td>&gt;1000</td>
</tr>
</tbody>
</table>

### 1.6 Requirements of a LHTES

A single PCM or a design of a LHTES cannot fulfill all the desired requisites. During designing a LHTES system, various requirements need to be addressed. These include: high energy density of the storage material (storage capacity), good heat transfers between the HTF and the storage material, mechanical and chemical stability of the storage material, compatibility between storage media and the container material, complete reversibility and the number of cycles, low thermal losses during the storage period, impact on environment and easy control [14]. Moreover, cost, operational strategy, maximum load and integration with other systems are also important.
1.7 Complexities with LHTES

As mentioned earlier, the LHTES has some excellent favorable criteria, but they have complex heat transfer aspects and have difficulty in tracking and predicting the solid-liquid interface during the phase change process making it a really challenging problem. These complexities and challenges can be summarized [14] as:

a) Nonlinearity in the phase transition resulting from the unpredictable motion of the solid-liquid interface during phase change,

b) Inadequate knowledge of the heat transfers within the liquid domain and across the solid-liquid interface

c) Unpredictable volume changes and structural behaviour within the PCM upon phase transition especially during solidification, such as formation of void/cavities, supercooling, nucleation etc. that may arise in the solid phase and can significantly reduce the heat transfer rate both during melting and solidification cycles

A good understanding on these aspects are of immense importance for LHTES system design perspectives. However, there is scarcity of available information in the literature concerning these issues.

1.8 Heat transfer associated with phase change

Phase transition resulting from this heat exchange is associated with various thermophysical changes in the PCM. Latent heat transfer problems are characterized by various enthalpic changes and significantly different from sensible heat problems since there lies a dynamic interface between the solid and the liquid phases, also sometimes called as a ‘moving boundary’ [34-36], across which thermophysical properties of the PCM vary drastically. The local behaviour of the PCM and the heat transfer process surrounding the interface is complex but crucial in defining the overall behaviour of the heat transfer process. Problems involving a phase change generally leads to a boundary that separates the two phases that are in thermal equilibrium, although it (interface boundary) marks an abrupt change between these two different structural entities. Any physical advancement
or retreat of this boundary into either phase can happen only if heat is absorbed or liberated from this interface region which may be considered as the ‘active zone’ [36].

Various mechanisms play important roles during phase transition, both for melting and solidification. The phase change process involves heat transfer, mass transfer (depending on the boundary conditions), absorption or release of latent heat, changes in thermo-physical properties of the PCM and, possible consequences of super cooling, nucleation, surface effects etc. [16]. A material, either in solid or liquid phase, is characterized by the strong presence of cohesive forces that keep the atoms in close proximity. In a solid phase, when the temperature is increased, the molecules in the material vibrate at an increasing rate maintaining a fixed position. In a liquid, since the molecules are comparatively loosely bonded than in solid form, besides vibration they can change their positions as well. The macroscopic manifestation of this vibrational energy is the heat or thermal energy and temperature is the measure of this phenomena [37]. Since the atoms in the liquid phase vibrates as well translates from its current positions, they are naturally more energetic than those in the solid phase when all other things being equal. Hence, before a solid material melts, it must gain a certain amount of energy to overcome the binding forces that are responsible to maintain the lattice of the solid structure. This particular energy is termed as the latent or hidden heat of fusion of the material denoted by $\Delta h_m$ [Ref] This latent heat represents the difference in thermal energy (enthalpy) levels between the liquid and solid phases, all other things being equal. In the opposite scenario, solidification of liquid requires the removal of this latent heat and the structuring of the atoms into more stable lattice positions [16]. All phase changes involve either an increase or decrease of intermolecular forces, thus the breaking or forming of intermolecular bonds signifies a phase change. In either case, there is a major re-arrangement of the entropy of the material, a characteristic of first-order phase transition [16].

Sometimes, there is a thin transitional zone between the distinct phases. When viewed in a molecular level, a phase change happens through the addition or subtraction of one molecular layer at a time, causing a mass transfer and is consistent with the way in which, many pure solid and liquid grow in saturated condition [38]. The interface formation may follow either of the two styles as described below. For strongly ordered crystalline solids,
the interface geometry usually follows the path that of the outline of the crystal structure. On the other hand, for a loosely ordered liquid, the interface geometry is influenced mostly by the surface tension of the liquid. For both the cases, although the structure of the interface, in general, is macroscopically smooth but there are presences of numerous edges and corners, which is one of the significant reasons behind its (interface) non-linearity [37].

1.9 Stefan problem: A classical 1-D heat transfer approach

Stefan problem is probably the simplest mathematical model describing the phenomenon of phase change where the heat transfer is considered unidirectional. When the phase change occurs, depending on the type of the PCM, an amount of latent heat per unit mass of the PCM is either absorbed or liberated with no appreciable change in its temperature. Few assumptions are considered in the Stefan problem [36):

1) No volume change in either phase during phase transition
2) Phase change temperature is constant
3) Each phase is homogeneous and isotropic
4) Heat transfer to liquid is only by conduction

To obtain a solution for the Stefan problem, the heat equation needs to be solved. Certain boundary conditions, called the ‘Stefan conditions’ on the evolving boundary is required to acquire a unique solution [36].

A simple schematic representation of the Stefan problem is shown in Figure 1.5. The solid-liquid interface is denoted as $x = s(t)$, where $x$ is the position in space, $s(t)$ is the free moving boundary and $t$ is the time. The phase change temperature is denoted as $T_m$, a fixed value assumed [16, 39]. Consider a domain $D$ with cylindrical cross-section containing two sub-domains $D_1$ and $D_2$ representing liquid and solid respectively. At time $t = t_0$ the sub-domains $D_1$ and $D_2$ are separated at $x = s(t_0)$, as seen in the figure. A uniform interface is assumed and the temperature $T$ depends only on $t$ and $x$.

In the case of the interface evolving to the right, when the solid is melting, it is expected that $T_{Liq} \geq T_m$ in the liquid phase and $T_{Sol} \leq T_m$ in the solid phase. At time $t=t_0$, the interface
has an area $S$. At time $t_1 > t_0$, the position of the interface shifted to a new position where $s(t_1) > s(t_0)$. Meanwhile, a volume of the PCM (cylindrical in shape) has melted which can be represented as $S*[s(t_1)-s(t_0)]$ and absorbed a quantity of heat $Q$, which is computed as $Q= S*[s(t_1)-s(t_0)]*\rho*\Delta h_m$, where $\rho$ and $\Delta h_m$ represents density and latent heat of the material, respectively [16, 39]. The heat must be provided by diffusion, and assuming no heat source or sink is present, by adopting the Fourier law of heat conduction, the heat flux can be calculated as [16, 39]:

$$q_{\text{Liq}}'' = -k_{\text{Liq}} D T_{\text{Liq}}$$

$$q_{\text{Sol}}'' = -k_{\text{Sol}} D T_{\text{Sol}}$$

$k$ represents the thermal conductivity of the material, $T_{\text{Liq}}$ and $T_{\text{Sol}}$ are the temperature of the solid and liquid.

![Figure 1.5: A PCM domain shown separately in two sub-domains with two phases, D1 (liquid) and D2 (solid)](image)

Some of the assumptions in Stefan problems are not realistic, especially the density which was assumed not to vary between phases. Also the heat transfer is neither one-dimensional nor through conduction in the liquid always.

1.10 Considerations of 2-D PCM storage geometry
In the present study, the PCM storage chamber was approximated as a two-dimensional geometry although in real applications all the storage chambers are three-dimensional. The reason behind this approach is to start with a simplified approach of the problem so that the analysis could be done easily and a comprehensive understanding can be achieved first before going into complicated geometries.

1.11 Convective heat transfer

Convection is the mode of heat transfer in the fluid (liquid or gas) domain. This heat transfer is dependent on the temperature difference between the bulk fluid and the fluid boundary adjacent to the heat source/sink, fluid thermo-physical properties and the behavior of the fluid flow.

Convective heat transfer coefficient, \( h \) is the most crucial parameter for a heat transfer problem in a liquid. ‘\( h \)’ is not a property of the fluid, rather a property of the fluid layer. It is an experimentally determined parameter, whose value depends on the type of the media (gas or liquid), their temperature dependent properties and all the variables that can influence convection such as:

\[
\dot{q}_{net} = Ah(T_{surf} - T_{\infty})
\]  

(1.6)
From the perspective of these discussions, it is clear that, to deal with any problem involving a phase change phenomenon, it is necessary to determine the instantaneous location of the interface, the manner and rate at which the solid-liquid interface is moving with time, the effects of input heat flux on the heat transfer process and most importantly, determining the convective heat transfer coefficient ($h$) in a transient manner during the phase change process. Knowing the changes in the thermophysical and chemical properties of the PCM over the system’s operating range and their effects on the heat transfer mechanism are also vital components.

1.12 Concerns with the thermal conductivity of PCM

Almost all phase change materials (PCMs), whether organic or inorganic, have a drawback of having low heat transfer rates during melting and freezing processes due to their inherent low thermal conductivity. Low thermal conductivity in PCM hinders the heat transfer process within its domain by prolonging the charging (heat addition) or discharging (heat rejection) period [40]. This problem not only drastically affects the melting and solidification performance of the LHTES system, but also limits their widespread use as latent heat storage material. The thermal conductivities of organic and inorganic materials usually swing around approximately 0.2 W/m.K and 0.5 W/m.K, respectively [41-43]. To have these PCMs successfully utilized commercially, some mechanism needs to be incorporated that can enhance the thermal conductivity. Enhancing the heat transfer rate through a PCM is a present day challenge.

Different approaches have been proposed to enhance the thermal conductivity of PCM. For example, the placement of metal structure (i.e. fins) inside a PCM [44, 45], use of porous material, [46-48], metal foams [49], carbon fibers [50] and dispersion of high thermal
conductivity nanoparticles in the PCM [51-59] are among the common techniques used to enhance the effective thermal conductivity of PCMs. Although there are many merits in these methods/applications, however, most of these enhancement techniques suffer from increased weight and volume in the system, except for the use of nanoparticles [60].

Nanoparticles possess physical and chemical properties, which are quite different from their bulk form [61]. Producing various types of nanoparticles has become cheaper due to rapid advancement in nanotechnology [62]. There is a growing use of nanotechnology in different engineering, electronics and industrial processes [63-65]. The presence of a mere amount of nanoparticles in the PCM significantly increases the effective thermal conductivity of the PCM, and consequently enhances the heat transfer characteristics. A fluid containing nanoparticles, generally less than 100 nm size is termed as nanofluid [66]. Nanofluids are prepared by dispersing such tiny particles (1-100 nm) [67] in a base fluid such as water, ethylene glycol, propylene glycol, oil and other conventional heat transfer fluids. Preserving the intended properties, holding nanoparticles in the fluid suspended after repetitive cycles and the cost of bulk volume of nanoparticles are the few major problems of widespread use of these nanoparticles.

1.13 Motivation

As discussed in the previous sections, thermal energy storages are expected to play a key role in the reduction of greenhouse gas emissions by increasing energy conservation and effective utilization of the clean thermal energy resources. Furthermore, it has been shown that the latent heat thermal energy storage has several advantages over other forms of thermal energy storage. However, due to continuous changes in solid and liquid phases along with their respective properties, the overall heat transfer process is very complex during the phase transition, which also varies with time. This is attributed to various complexities such as, the non-linearity of the solid-liquid interface movement and its dynamics, due to the continuous changing of the volume fractions of solid and liquid. The complexity further arises due to the geometry and orientation of the heat source, which heavily influence the dynamics of the interface movement.
The literature review revealed that most of the studies on convective heat transfer problem were focused on single phase fluids and very limited studies investigated heat transfer during the course of phase change process. Furthermore, majority of the studies used simplified numerical approaches and limited number of simple configurations of the PCM domain and heat source orientation.

However, there is a lack of thorough understanding of the two-phase heat transfer process considering the transient nonlinear dynamics of the solid-liquid interface. There is also scarcity of studies directly investigating the phase change process and analyzing the local variations in the convective heat transfer coefficient and its impact on flow characteristics. Therefore, a detailed study on the phase change process in a transient manner and to understand all related thermophysical parameters and their correlation with the solid-liquid interface characteristics is vital to improve understanding of the heat transfer process in a LHTES. This would help to improve and optimize the TES system design for its utilization in various applications for sustainability. Although the use of nanoparticles as thermal conductive enhancers has been studied to a good extent but, mostly these are with single fluid or a mixture of fluids. There is a lack of detailed studies on their behaviour and performance with phase change materials.

While it is important to investigate and characterize transient heat transfer process over the entire cycle i.e. melting (charging) and solidification (discharging) processes, due to the extensiveness of this task and various experimental challenges, the present research is focused on the detailed investigation of the transient heat transfer process during the melting phase only, which itself is a comprehensive task.

1.14 Objectives

The objectives of this experimental study are to:

a) Conduct a detailed investigation of the heat transfer phenomena in a PCM during solid-liquid phase change (melting) process with different heat source configurations
b) Investigate the effects of thermal conductivity enhancers during the heat transfer process in a transient manner in a PCM.

The focus of the present research work is on the detailed investigation of the underlying fundamental processes that influence the solid-liquid phase change and their characterization. The work is conducted through experimental investigations using Paraffin wax as the PCM. Being focused on the fundamental aspects, the outcome of this research work is expected to improve understanding of the heat transfer process during the phase change and its predictions. This will be useful in further modelling and designing of thermal energy storage systems.

In this study, the heat transfer mechanism in a phase change material involved in a solid-liquid phase transition process was experimentally investigated with two very fundamental and classical design of the heat sources that are commonly encountered in practical field. A horizontal cylindrical heat source near the bottom of a rectangular shaped PCM storage chamber was used in the first study. For the second study, a U-shaped vertically oriented cylindrical heat source was used in a similar chamber. Temperature measurement and image acquisition systems were employed in order to gather data from different experimental runs and processed to generate various useful thermo-physical parameters during melting of a PCM.
1.15 Thesis layout

**First chapter** introduces the broader scope of the present research by describing the role of thermal energy storage in the context of energy and sustainability. It then identifies different types of thermal energy storages and focuses on the detailed description of the latent heat thermal energy storage and associated processes. The current challenges and issues related to the latent heat thermal energy storage are discussed next. The motivation and the objectives of this present research are then provided. At the end, the methodologies and a brief description of the techniques used for the measurements and data processing that are used during this study are presented.

**In second chapter** an experimental investigation is reported for a solid-liquid phase change process in a thermal energy storage chamber having a cylindrical heat source near the bottom of the chamber. An image-based analysis is used to investigate and describe the interface development and fluid flow behaviour. Detailed results on the effect of changes in the input heat flux and various thermo-physical parameters are presented and discussed.

**Third chapter** focuses on the investigation of solid-liquid phase change process in a thermal energy storage chamber with a U-shaped cylindrical heat source to better understand the effect of the geometry and orientation of the heat source on the phase change and heat transfer behaviour. Detailed results on the effect of changes in the input heat flux and various thermo-physical parameters are presented and discussed.

**Fourth chapter** focuses on the influence of thermal conductive enhancers (TCE) in the PCM on the solid-liquid phase change process and associated heat transfer. The same heat source configuration as in Chapter 2 was considered. A detailed comparative study is presented to see the effects of the TCE on the heat transfer parameters of the PCM.

**Fifth chapter** summarizes the conclusions of each chapter and presents an understanding of the heat transfer phenomena in a PCM based thermal energy storage system and lists some future recommendations.
References


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Chapter 2

2 Characterization of the melting behaviour and the transient heat transfer in a phase change material

2.1 Abstract

An experimental study is reported that investigated the melting and associated heat transfer in a phase change material (PCM) domain in a transient manner. The experiments were conducted in a rectangular chamber with a horizontal heat source located near the bottom of the chamber. The spatio-temporal behavior of the melting process and the temperature field were obtained through simultaneous imaging and temperature measurements. Results show a strong dependency of the solid-liquid interfacial movement on the melting process that transitioned from linear to nonlinear form with the growth of the melted fraction due to the presence of convective cells. The interfacial velocity magnitude was also found to increase with an increase in the input heat flux. The transient heat transfer coefficient was also computed. The results show that the convective heat transfer coefficient increased rapidly in the early stage of melting and became nearly steady once the liquid fraction became dominant. The evolution and growth of the convective cells in the liquid PCM domain was attributed to this trend. A new Nusselt-Rayleigh numbers correlation is proposed to characterize the heat transfer during the melting phase.

Keywords

Thermal energy storage, PCM, latent heat, natural convection, solid-liquid interface, convective heat transfer coefficient

1 The chapter is a journal article submitted to the International Journal of Heat and Mass Transfer
2.2 Introduction

Demand for the energy is growing at a phenomenal rate. To meet the growing energy demand, more than 80% of total energy is provided by fossil fuels [1]. This trend has two major consequences; a rapid depletion of conventional fossil fuel reserves and an increase in greenhouse gas (GHG) emissions that are detrimental to the living environment. While various renewable energy resources have a significant potential to contribute to this energy demand, certain geo-physical constraints such as unpredictable availability, seasonal variations etc. hamper their effective integration into the energy mix and often due to the mismatch between the demand and supply.

Other important factor that affects the sustainable energy consumption is the wasted heat in industrial, commercial and residential sectors because of the inefficiencies in the processes as well as in the utilization patterns. It has been estimated that over one half of the input energy in the industrial sector is discarded as waste heat with the flue gases or through other forms of low-grade energy [2]. If all of these wasted energies are accumulated, it is not surprising to say that, they would come up as another massive source of usable energy, if they can be reused intelligently in an efficient, feasible and effective manner.

A prospective way to address the issues of mismatch in supply and demand of thermal energy from renewable resources as well as the recapturing of waste heat, would be to store thermal energy in a suitable form during its availability and utilize it during the periods of demand. Thermal energy storage (TES) is basically a temporary storage of heat energy for its intended use at a later time. Heat transfer is in general a rate determining process [3]. The duration of the energy storage depends on the requirements of its use which dictates the storage system design. Thermal energy can be stored generally in the form of sensible heat, latent heat or thermochemical energy [4]. Thermal energy storage (TES) is by far the most effective and practical solution to address these issues and as a result, has the capability in providing efficient means for energy management [5]. TES is basically an enclosure containing some suitable material(s) as a medium to store thermal energy, taking advantage of some of the thermo-physical properties of that material(s). The available
thermal energy is routed to the TES via a suitable medium (usually a fluid) where the heat transfer takes place and the storage material absorbs and holds the thermal energy. When required, the same phenomenon is reversed and the thermal energy is extracted from the storage material.

As regards to the criteria for selecting a suitable storage medium, the phase change materials (PCMs) are very attractive candidates in a thermal energy storage system and has drawn huge interest in recent years. The criteria include but not limited to the thermo-physical benefits, environmental effects and economy. The use of PCMs as storage media is by far the most effective and efficient way to storing thermal energy by exploiting the enthalpy of phase change as the storage mechanism [6]. Among all possible phase change processes, the solid-liquid phase change is considered as the most suitable form for TES applications [7-8]. The use of PCM in TES has many advantages over other forms of energy storage mediums, especially the sensible heat storage. Because of PCM’s high energy density (amount of heat energy storage per unit mass), it is possible to have much smaller and compact storage facilities compared to sensible heat storage types [9]. The characteristic ability of PCM’s narrow temperature range during thermal energy charging and discharging periods has made it a suitable candidate for applications where smaller range of temperature variations are important.

Heat transfer processes involving solid-liquid phase transformation have been an attractive subject and has drawn continued interest for several decades by scientists and researchers. The reason most likely is driven by the enormous practical applications associated with this phenomenon. These studies occur in a diverse range of applications, such as metal processing, building systems/environmental engineering, electronics cooling, food processing and preservations, spacecraft thermal applications and thermal energy storage system to name a few [4, 10-14].

When a substance undergoes a phase transition due to heat transfer, the exchange process between the phases occurs in the interfacial region that separates both solid and liquid domains. This region is termed in the literature as the active zone [15]. The dynamics of the solid-liquid interface, also called a ‘moving boundary’, [3, 15-16] plays a very crucial
role in regulating the transport of heat across the interface. The complexity of the problem arises due to the increase in the volume fraction of one phase simultaneously with a decrease in the volume fraction of the other phase. This behavior has a significant impact on the mode of heat transfer within a particular phase. The solid-liquid interface not only physically separates the two phases but also isolates these two regions that have significantly different thermo-physical properties. The thermos-physical properties vary considerably between phases, which result in the transport of energy from one phase to other in a complex and transient manner. Viskanta [15] argued that in such cases, the position of the moving boundary cannot be identified in advance; however, its determination is an important constituent of the solution. The energy absorbed or released in this active interfacial zone determines the rate and manner at which the interface would move and/or develop.

To analyze this transient heat transfer process from theoretical and computational aspects, the conduction in solid phase needs to be coupled with the convective heat transfer in the adjacent liquid phase, and both are required to be analyzed separately and then conjugated [17,18]. The coupling and non-linearity of the governing equations and the complicacy in geometries involved by the moving interface boundary put constraints on analytical and numerical approaches to study this problem and hence, the experimental tools play a very vital role to investigate the problem [19, 20]. Accurate prediction of the convection heat transfer is one of the most critical components in designing a TES.

Since melting and solidification in thermal energy storage is a cyclic process and require to continue over a long period of time, the nature of the solidification of PCM in every cycle has enormous effects on the consecutive melting sequence [21]. The interfacial morphology and microstructure of the material developed during this solidification stage largely affect the interface velocity, melted fraction, temperature gradient and the melting time in the following melting sequence [15, 20, 22]. The melting time of a PCM is also an important factor in determining the performance of a thermal storage system since it is related to the rate of heat absorption by the PCM [23, 24].
The natural convection during melting of a PCM in an enclosure is a critical factor and plays the most crucial role. Hale and Viskanta [25] conducted an experimental investigation to understand the effects of convection on the melting of paraffin (n-octadecane) inside a rectangular geometry heated from a side wall. They observed the shape of the solid-liquid interface, temperature profiles and heat transfer coefficient and reported the significance of natural convection that needs to be studied further. Ho and Viskanta [16] experimentally and numerically studied melting of the wax inside a rectangular cavity and reported that except for the initial stages of the melting process, the natural convection significantly affected the melting and became the dominant mechanism after a short duration of conduction dominated period. They argued that the natural convection is actually the key factor in controlling the melting rate and shape of the solid-liquid interface. Gadgil and Gobin [26] argued that since convection is the dominant mode of heat transfer during a melting process compared to conduction, the latter can be ignored in the analysis of the heat transfer process during melting.

Gau and Viskanta [27] experimentally investigated the impact of natural convection on the solid-liquid interface motion and heat transfer mechanism during the melting and solidification processes of pure gallium inside a rectangular enclosure when heated from a side wall. They measured the position of the interface and the temperature distributions. The solid-liquid interface positions and the fluctuations observed in the temperature measurements were utilized as qualitative and quantitative indicators of the convective flow regimes and to understand the structure of the flow in the melted domain during phase transition. For the case of melting, they correlated the melt volume with heat transfer coefficients through relevant dimensionless parameters, i.e. Nusselt number ($Nu$), Rayleigh number ($Ra$) and Stefan number ($Ste$). Rayleigh number represents the ratio of the buoyancy force to the viscous force in the liquid domain, Nusselt number is a ratio between the convective and conductive heat transfer and the Stefan number represents the sensible heat to latent heat ratio in the storage medium. Cristiano and Luiz [11] numerically studied the melting process of Tin in a 2D rectangular enclosure heated through a side wall, while the other walls were kept adiabatic. They found that an increase in Rayleigh number and Stefan number reduced the total melting time, it was apprehended
that the increase in Rayleigh number enhanced natural convection in the system resulting in higher melting rate. Similar consequence was observed for the Stefan number.

For natural convection in rectangular enclosed geometries, a core region exists outside the surrounding boundary layers that are formed at the enclosure walls. Basak et al. [28] reported that the interactions between the surrounding boundary layers and core flow create complications in the analysis of the natural convection problems. As a result, the local and bulk parameters also varies.

Most studies on melting and phase transition reported in the literature considered the side wall of the enclosure as the heat source [27, 29-33]. There is a scarcity of studies on the phase change process inside an enclosure that considered heating from the bottom [4, 24, 34, 35]. The natural convection process and the dynamics of the interface movement change substantially for the bottom heat source versus the side heat source. This scarcity is likely due to the difficulty of predicting and understanding the non-linear nature of the interface movement, interaction between solid and liquid phases as well as the complex nature of the heat transfer between the phases [4, 10, 24, 36].

Fluid flow due to the natural convection around a horizontal cylinder is observed in many practical engineering applications and is considered as a classical reference case for numerical model validation. Horizontal cylinder acts as a line heat source when the Rayleigh number is small but at high Rayleigh numbers, a distinct boundary layer is formed around the cylinder which is of interest for heat transfer analysis. Although some studies can be found in the literature, but there exists a wide dispersion in the analytical, numerical, and experimental data. The reasons behind the differences observed especially in analytical and numerical methods are due to many assumptions considered to simplify the problems, for example, simplified boundary conditions, domain size and the use of simplified boundary layer equations. Natural convection experiments are very complicated to analyze as it is very difficult to isolate the natural convection phenomena from other influencing factors like, loss due to heat conduction, effects of radiation, shape and size of the experimental chamber, location and orientation of the heat sources and predicting the effects of external temperature and fluid velocity [37].
Gau et al. [38] visualized the melting process of paraffin wax in a rectangular enclosure heated from below and reported that the buoyancy-driven convective flow pattern dictates the solid-liquid interface shape, similar to other researchers. This work was further extended by Diaz and Viskanta [39] to study the solid-liquid interface behavior during n-octadecane’s melting process. It was observed that the initiation of natural convection was depended on the Rayleigh number. They also observed the development of multiple Benard-like small sized convection cell during the initial stage of melting, all of which grew larger with time and subsequently merged with neighboring cells to form a single cell. They also argued that the effect of initial sub-cooling of the solid was the reason for the delay in the development of the solid-liquid interface.

Prud’Homme et al. [40] conducted numerical studies in a vertical rectangular enclosure where the heat source was at the bottom and the side walls were assumed adiabatic. They observed the onset of convection at a critical Rayleigh number of 2197, which they argued was based on the melt layer thickness. They also reported that the initial multicellular convective cells in the liquid domain were ultimately converted to a single Bénard cell as the melted zone grew. Jellouli et al. [19] numerically studied the heat transfer problem in a rectangular PCM based storage enclosure that had a bottom heat source. They argued that the growth of instabilities of Rayleigh-Benard convection along with the moving boundary, increases the complexity of the problem. Hsu et al. [41] conducted numerical simulations in a rectangular domain with a cylindrical heat source at the bottom. They considered ‘pseudo-convection’ and argued that the velocity of the interface was very small compared to the bulk convective flow and hence, the convective flow was not influenced by the interface motion. Benard et al. [42] presented results on the melting process of PCM inside a rectangular enclosure where conduction in the solid and natural convection in the liquid phase were considered as heat transfer modes. They observed that the heat transferred by conduction in the solid phase significantly modified the dynamics of the melting process. Yingxin et al. [12] studied through numerical investigations of the melting of ice from a bottom heat source, reported that the heat transfer rate in the liquid domain is much larger than the heat transferred in the solid domain. They indicated that the duration of the melting period of the PCM is a more precise way to quantify the heat transfer rather than quantifying the temperature difference of the HTF fluid at the inlet and exit. Recently, Feng
et al. [4] investigated the melting of a PCM in a bottom heated rectangular cavity. They also found that conduction is the dominating mode of heat transfer during the early stages. They also argued, when the convection heat transfer was evolved, convective cells were observed and the solid-liquid interface was found to be parallel to the bottom wall. As convective cells grew and intensified, they observed the deformation of the interface.

As the above literature review shows, although significant work has been conducted to investigate the solid-liquid phase change process, there is a scarcity of thorough studies characterizing the transient heat transfer associated with the spatio-temporal variation of the solid and liquid phases during solid-liquid phase change process. A better understanding of this transient heat transfer process is vital to improve the design and performance of phase-change heat storage systems, especially the cycle time, the heat transfer rate, and thermal efficiencies [20].

The present study is comprised of experimental investigation to improve understanding of the underlying processes during solid-liquid phase transition. The specific focus is on the investigation of the transient solid-liquid interface movement and the associated convection heat transfer in the melted region.

2.3 Experimental setup

2.3.1 Storage Chamber

The schematic and photograph of the experimental setup are shown in Fig. 2.1 (a) and (b), respectively. The thermal storage is a rectangular shaped chamber with the inside cavity dimensions of 100 mm long (L), 100 mm high (H) and 12 mm deep (D) as shown in Fig. 2.2 (a) and (b). The main chamber was made from an 8 mm thick optically clear cast acrylic plate cut into a U-shaped configuration. A plain clear glass pane and an acrylic sheet each of size 150 mm × 125 mm × 5 mm were used as the front and back walls of the chamber, respectively. Two, 2 mm thick rubber gaskets were sandwiched all along the contact area between the main chamber and the front and back walls to prevented leaks and minimized heat loss to the surroundings.
A copper tube was inserted horizontally near the bottom of the storage chamber via two holes in the acrylic plate. The copper tube had an outer diameter (OD) of 4.76 mm (3/16") and a wall thickness of 0.355 mm (0.014"). Hot air passing through this metallic tube acted as the heat source throughout this study. The total length of the heat transfer tube in contact with the PCM inside the chamber was 100 mm. Rubber seals were used between the tube and the chamber walls to prevent a direct contact of the tube with the acrylic block. These rubber seals not only minimized the heat transfer between the tube and the acrylic plate, but also restricted any leakage of the liquid PCM from the storage chamber. All parts of the setup except the front glass pane were insulated to avoid heat losses. The storage chamber was attached onto a metal frame. The upper end of the chamber was left open to room atmosphere to accommodate thermal expansion of the PCM during melting and with the anticipation that it would indirectly help in the fluid flow and heat transfer process by minimizing the pressure buildup due to volume expansion of the PCM [36]. In a real thermal storage, heat is continuously transferred from the heat source towards the boundaries of the storage domain. Hence, the heat transfers through the open top prevented the heat buildup in the chamber and hence mimicked the condition of a continuous heat transfer in a real thermal storage.

The use of clear glass pane allowed optical access to facilitate imaging of the melting process inside the chamber. Since the front glass pane was not insulated, some heat loss occurred through this pane. To ensure that this heat loss did not affect the underlying melting process and associated heat transfer within the PCM domain, a separate set of experiments was conducted by insulating the front pane under the same conditions (see appendix B for location of the insulation). The temperature data were recorded and compared with the corresponding cases with uninsulated front glass pane. The results show that the PCM temperature within the domain for fully insulated case was slightly higher that the case with the front glass pane, as expected. Comparison of results at different locations within the domain shows that the difference in the mean temperatures for the two cases is on average 8.7%. Due to the transient nature of the process, the
Figure 2.1 (a): Schematic diagram of the experimental setup. The horizontal heat source (HTF tube) is shown near the bottom of the chamber.

Figure 2.1 (b): Photograph of the experimental setup. The horizontal heat source (HTF tube) is shown near the bottom of the chamber.
Figure 2.2: Schematic of the PCM chamber, (a) Isometric view, (b) locations of thermocouples marked with open circles
temperature profiles at different locations within the domain over the entire melting period were compared for both the cases. The results show almost similar trends for both insulated and uninsulated cases confirming that the uninsulated pane did not have any considerable influence on the underlying transient melting process.

2.3.2 Heat source

As mentioned earlier, controlled hot air passing through a horizontal copper tube inside the PCM domain acted as the heat source. The air (hereinafter termed as the heat transfer fluid, HTF) was heated by an air heater (Zesta ZA0017) equipped with a PID controller (Zesta ZEL-9100) to control the temperature of the HTF fluid. The accuracy of the PID controller was within ±2°C. To minimize pressure fluctuations of incoming air, the compressed air from the supply line was first passed through a settling tank to dampen any pressure fluctuations. A pressure regulator controlled the pressure of the air exiting the settling tank. The air was then passed through a flow meter to control its flow rate before passing through the heater (figure 2.1(a). The flow rate of air was kept between 30-40 liters/min. The accuracy of the flow meter was estimated as ±5%. Four different input heat loads were applied by varying the HTF inlet temperature to 125 °C, 150 °C, 175 °C and 200 °C respectively through the temperature setting in the heater controller. The average heat flux for each experiment was computed as 6070 W/m², 8150 W/m², 10,030 W/m² and 11,475 W/m² respectively.

2.3.3 Phase Change Material (PCM)

Polyfin, a mixture of fine paraffin wax and copolymer alloys in the form of milk-white pellets was used as the phase change material (PCM) in this study to take advantage of its clear appearance upon melting. A small amount of tracer particle (silver coated hollow glass sphere with an average diameter of 9-13µm) was added to the wax to observe the flow behavior of the molten liquid through imaging. The volume change of the wax in the liquid stage was estimated to be approximately 10% from the solid state. Thermo-physical properties of the PCM is shown in Table 2.1.
Table 2.1: Thermo-physical properties of PCM at 25 °C (VWR CA 27900268) [33-34]

<table>
<thead>
<tr>
<th>Melting Temp., °C</th>
<th>Specific gravity</th>
<th>Specific heat, $C_p$ kJ/kg/K</th>
<th>Thermal conductivity $k$, W/m/K</th>
<th>Kinematic viscosity $\nu$, m²/s</th>
<th>Latent heat $\Delta h_m$, kJ/kg</th>
<th>Thermal diffusivity $\alpha$, m²/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>0.92</td>
<td>2.890</td>
<td>Solid ~0.24</td>
<td>Liquid ~0.18</td>
<td>5.50×10⁻⁶</td>
<td>180</td>
</tr>
</tbody>
</table>

2.3.4 Temperature data acquisition

A total of 31 $T$-type thermocouples (T30-2-507, Zesta Engineering Ltd., hereinafter referred to as TC) were used to measure the temperatures at various locations in the experimental domain. These locations are marked as circles and shown in figure 2.2(b). Two thermocouples were exclusively employed to measure the temperatures of air (HTF) at the inlet and outlet of the PCM chamber. The temperature on the surface of the HTF tube inside the chamber was measured at five different locations along the tube length with equal spacing of about 20 mm between them. The first and last thermocouples on the tube surface were located 8 mm from the chamber inlet and outlet, respectively. For temperature measurements within the PCM domain, 20 thermocouples were arranged in an array format of 4 rows and 5 columns (figure 2.2(a) and (b). The distance between these thermocouples was also kept at about 20 mm while, the first row was located 15 mm above the HTF tube. To measure the temperature of the PCM very close to the HTF tube surface, four more thermocouples were placed at 1, 2.5, 5 and 10 mm above the HTF tube surface at the mid axial location (figure 2.2(a) and (b). All thermocouples were inserted into the chamber through the insulated acrylic back wall.

The temperature data from all thermocouples were recorded simultaneously via data acquisition cards (thermocouple modules: NI-9214 and NI-9211) controlled by the LabVIEW software at a rate of 3 Hz. The temperature data recording was started 1 min prior to the beginning of the heating process and continued 5 min after the PCM was completely melted.
2.3.5 Thermocouple Calibration

All thermocouples were calibrated by placing them in a well-insulated hot water bath along with a high accuracy thermometer (Kessler 10/30 ST Jointed Thermometer) which was used as the reference. The calibration was conducted over a temperature range from 90 °C to 0 °C with approx. 10 °C temperature increments. Calibration curves showed almost linear relationship. The uncertainty of thermocouples was ±0.4%.

2.3.6 Computation of the heat input to the PCM

The total heat loss by the HTF passing through the chamber was calculated from the steady-flow thermal energy equation [43]:

\[ \dot{q}_{\text{total}} = \dot{m} c_p (T_{in} - T_{exit}) \]  

where, \( \dot{q}_{\text{total}} \) is the heat transfer rate, \( \dot{m} \) is the mass flow rate of air, \( c_p \) is the specific heat of air, and \( T_{in} \) and \( T_{exit} \) are the inlet and exit temperatures. The net heat transferred (\( \dot{q}_{\text{net}} \)) to the PCM is then calculated by subtracting the heat loss from the HTF tube between the chamber inlet and outlet and the corresponding thermocouple locations, from the total heat loss by the HTF. The uncertainty in estimating the heat flux was on average within ±10% (see appendix C).

2.3.7 Image acquisition

A high resolution 4 Mega Pixel CCD camera (Flare 4M180-CL) with the resolution of 2048 × 2048 pixels was used to capture the images of the phase change process and track the solid-liquid interface and the flow behavior of the melted wax. The camera was connected to an image acquisition system (DVR express core, IO Industries) to record images. The camera was controlled via the DVR express CoreView software installed on a PC. The field of view of the camera was set at 94 mm × 94 mm. For each run, images were captured from the beginning of the melting process until the PCM was completely melted. The camera frame rate was set between 30 and 40 frames per second for different experimental runs depending on the input heat flux.
2.3.8 Interface detection

The solid-liquid interface was detected in each image using an in-house Matlab code. The code implemented image processing routines to detect the interface based on the change in gray-scale values between solid and liquid phases. A series of morphological operations were performed to remove noise from the data. The coordinates of the detected interface were recorded at every spatial location. Fig. 2.3 shows an original image where white region represents the solid phase and dark gray region represents the melted wax. The interface detected by the in-house code is also plotted for comparison. The uncertainty in detecting the interface is within ±2 pixels that corresponds to ±90 μm. By processing a series of consecutive images from the beginning to the end of the melting process, a time history of the interface movement was obtained. This information was used in the further analysis of data. Note that the interface movement in the immediate vicinity of a thermocouple was locally disturbed. Hence, the interface data in the vicinity of each thermocouple were excluded from subsequent analyses. The red line shows the detected solid-liquid interface. The thermocouples are also evident in the melted domain. The heat transfer tube is shown at the bottom of the image.

Figure 2.3: Instantaneous image showing the PCM chamber domain during melting. Dark grey and white regions represent liquid and solid phases respectively.
2.4 Results

The results obtained from the experiments were analyzed and presented from different aspects. The placement of 24 thermocouples in the PCM domain along with the imaging, provided a better insight into the thermal and dynamic behavior of the solid-liquid phase change process.

2.4.1 Spatio-temporal behavior of the PCM during melting

Figure 2.4 shows a sequence of temperature fields from thermocouples and the corresponding images to illustrate the PCM behavior inside the chamber during the phase change (melting) process. In the initial stages of heat transfer to the PCM, the temperature of the PCM in the vicinity of the HTF tube increases through conduction. Once the melting point is reached, the phase change starts and a thin layer of liquid PCM is formed around the tube. Since the HTF continuously loses heat to the PCM, the temperature of the HTF decreases along the tube, which causes a decrease in the tube surface temperature (figure 2.4(a). As a result, the local heat transfer rate from the tube to the PCM decreases along the tube length. Thus, more heat is transferred to the PCM near the tube inlet and hence, the melting process initiates in this region. In the early stages of melting this non-uniformity in the heat transfer rate is more prominent. As the image in figure 2.4(a) shows, thickness of the melted PCM layer is largest above the tube near the inlet, which gradually decreases towards the outlet. The higher PCM temperature is observed in the melted region, while the solid region shows a uniform low temperature. As the melting further progressed with time, the melted fraction continued to grow upwards and the temperature of the melted PCM continued to rise. The non-uniformity in the heat transfer rate causes a non-uniformity in the liquid PCM particularly in the vicinity of the solid-liquid interface. As figure 2.4 (b-c) shows, the temperature of the liquid PCM near the heat source inlet of the chamber is higher, which gradually decreased towards the outlet. This effect however, is more profound when the melted PCM fraction is relatively low and not far from the tube surface. As melting process continued, the temperature in the bulk melted PCM domain tend to become more uniform both axially and vertically, however, strong vertical temperature gradients remained in
Figure 2.4: (Left) Colormaps of the temperature field in the PCM chamber at different stages of melting. The color blocks in the bottom row represent the surface temperature of HTF tube. (Right) Corresponding image of PCM domain.
the vicinity of the interface (figure 2.4 d-e). Once the PCM is completely melted, the bulk temperature remained uniform (figure 2.4 f). The image sequence in figure 2.4 shows that the interface is relatively linear in the early stage of the melting process although it is inclined due to the higher melting rate near the inlet which gradually decreased axially. As the melted fraction grows, the interface profile started to show nonlinear trends. This is likely due to the formation of convective cells within the liquid domain that locally change the transportation of heat and hence the local melting rate.

2.4.2 Interface dynamics

The image sequence in figure 4 shows the dynamics of the solid-liquid interface during the melting process. To quantify this behavior, the detected solid-liquid interfaces from a sequence of images describing the time history of the interface progression were analyzed. The spatio-temporal data of the interface position allowed to compute interface velocity as a function of time and space. Figure 2.5 (left side) shows the spatio-temporal variation of the solid-liquid interface during melting, at four heat fluxes. The time interval between the successive interface profiles was 133.33 sec for all presented cases. The right side in figure 2.5 shows the time history of the local velocity at four axial locations within the PCM domain. At the lowest heat flux (figure 2.5a), the interface movement in general was relatively uniform over time, although local spatial variations are visible in the plot. Some local small-scale variations due to the presence of thermocouples are also evident. However, as the plot shows, their effect was very local, which did not induce any large scale spatial or temporal variations in the interface. The plot also shows some local acceleration and deceleration of the interface, with a prominent acceleration in the region at a distance of 5-7 cm across the chamber at a height of about 3 cm. This local acceleration/deceleration could be due to the compactness or microstructure of the local wax, for example, presence of void or super cooling during solidification process [15, 20, 22]. At a later stage of melting, the interface profile acquired an overall nonlinear form.

As the heat flux increased, the local fluctuations in the interface grew mostly in the later stage of melting (figure 2.5b). The transformation of the interface profile from linear to
Figure 2.5: (Left) The time history of the solid-liquid interface position, (Right) the corresponding interface velocity profiles at different axial locations. The results are presented for four input heat fluxes, (a) 6070 W/m², (b) 8150 W/m², (c) 10,030 W/m² and (d) 11,475 W/m²
nonlinear form with the advancement of melting is also evident. As the heat flux further increased, the results show that the local acceleration of the interface increased not only in magnitude but also in the spatial and temporal extents (figure 2.5c). Similar to the previous case, this acceleration occurred in the later stage of melting. The extent of this acceleration indicates that it is not due to the local compactness of the wax as observed at low heat flux cases. A detailed analysis of the image sequence which also show the tracer particles’ movement, revealed that the convective cells formed within the melted PCM domain locally enhanced heat transfer at the interface, which caused a faster advancement of the local interface resulting in a rapid enhancement of local acceleration. For this particular case, the results show that the convective cell on the right side of the chamber is stronger than that on the left side. The plot also shows that these local convective cells significantly altered the interface profile making its shape complex and highly nonlinear. A similar trend is observed when heat flux was further increased. Comparison of results in figure 2.5 (c) and (d) shows that the influence of convective cells on the local interface movement becomes significant when more than half of the PCM in the domain transitions to the liquid phase. It is also observed that this influence diminishes when the melting process approaches the final stage i.e. the interface reaches near the PCM domain boundary.

Figure 2.6(a) shows the spatially-averaged interface velocity as a function of time for all four heat flux cases. As observed, at the lowest heat flux, the interface velocity remained relatively uniform. This could be due to the reason that the local acceleration and deceleration of the interface was minimal (figure 2.5a). Similar behavior is observed for the next higher heat flux (8150 W/m²) although the overall velocity magnitude increased with an increase in the heat flux. At the two highest heat fluxes, large variations were observed in particular during the latter half of melting, which are likely due to the presence of strong convective cells.

The interface velocity was in general higher for the higher heat flux case. The results also show that the increase in the interface velocity from the heat flux of 10030 W/m² to 11475 W/m² is percentagewise not as large as that at lower heat fluxes. This indicates that the local convective cells when present, plays a dominant role in controlling the
Figure 2.6: (a) The spatially-averaged interface velocity as a function of time for four heat flux cases. (b) The time-averaged interface velocity over the spatial extent (across the chamber distance) of the PCM domain at four heat fluxes.
Figure 2.6: (c) Spatio-temporally averaged interface velocity versus the input heat fluxes. (d) Spatio-temporally averaged interface velocity versus Stefan number (Ste)
interface velocity. Results in figure 2.6 (a) also show that in general, the interface velocity decreases in the initial stages of melting. Diaz and Viskanta [39] also observed this trend during melting of n-octadecane with a bottom heat source. They argued that the decrease in the interfacial velocity in the early stage of melting is due to the delay in the development of solid-liquid interface morphology that is attributed to initial subcooling. The results also show a decrease in the interface velocity in the final stage of melting particularly at the two higher heat fluxes. This trend can be explained based on the presence of the convective cells. As mentioned earlier, the convective cells locally increase the melting rate and hence accelerate the interface movement. In the final stage of the melting, the interface reaches close to the upper edge of the PCM domain boundary where the role of the convective cell diminishes. This effect is clearly visible in figure 2.5 (c & d).

Figure 2.6(b) shows the time-averaged interface velocity over the spatial extent of the PCM domain at different heat fluxes. As observed, the interface velocity at the two lower heat fluxes varied nonlinearly over the spatial extent of the chamber. At both cases, the interface velocity was relatively higher in the left half of the PCM chamber, which is due to the reason that the heat transfer fluid (HTF) entered from the left end of the PCM chamber and hence higher heat transfer rate is expected in this region due to higher HTF temperature. At the two higher heat flux cases where the convective cells were fully established, the local interface velocity magnitude was found to be significantly influenced by these convective cells. It is found that these convective cells could increase the local interface velocity magnitude by approximately 25%. The spatio-temporal averaged interface velocity for all heat flux cases is shown in figure 2.6(c). The results show an increase in the interface velocity with heat flux until the case when strong convective cells emerged. Once the convective cells were formed, the results show no significant change in the interface velocity with a further increase in the heat flux. Stefan number (Ste) is a non-dimensional parameter that describes the ratio between the sensible and latent heats of the substance [15]. In the present study, the Stefan number increased with an increase in the heat flux due to the increase in the temperature difference which implies that the sensible heat fraction increased with the heat flux. The spatio-temporal averaged interface velocity is plotted versus the Stefan number in figure 2.6(d). The results show that the interface velocity increased almost linearly with an increase in the Stefan number.
2.4.3 Temperature profiles

As mentioned earlier in the experimental setup section, temperature was measured at 24 locations inside the chamber to characterize the thermal behavior of the PCM during the melting phase. The time history of the normalized PCM temperature at eight vertical locations (1.0, 2.5, 5, 10, 15, 35, 55 and 75 mm from the top surface of the HTF tube) in the mid-axial plane of the PCM domain are plotted in Figures 2.7 at different input heat flux conditions. In the normalized form, $T_m$ corresponds to the melting temperature of the PCM and $T_s$ corresponds to the HTF tube surface temperature. Figure 2.7(a) shows the PCM temperatures at the lowest heat flux of 6070 W/m². As the figure shows, at the beginning of the melting process, the temperature along the height of the PCM domain was the same, equal to the room temperature. As the melting progressed, the rise in the PCM temperature was strongly dependent on the vertical distance from the heat source. At a location closest to the heat source (i.e. 1 mm away), the PCM temperature sharply rose and quickly passed the melting temperature of the PCM, as expected. At greater heights, the heat initially diffused through the solid phase via conduction and eventually the solid PCM passed the melting stage into the liquid phase. The rate of heat diffusion is correlated with the rate of temperature rise, hence, as expected, the heat diffusion rate decreased with an increase in the distance from the heat source. The results at other higher heat fluxes (figure 2.7 b-d) also show similar behavior.

The plots in figure 2.7 also show that at a given location, after reaching a certain temperature that is above the melting temperature, the PCM temperature reached a quasi-steady state. The corresponding interface position data shows that this state was reached when the PCM at that thermocouple location was fully melted. This observation helps in providing the physical insight into the melting process of a PCM, which is described as follows. As the heat transfer begins, a given mass of PCM in the solid phase receives heat from the adjacent PCM mass with higher energy that is closer to the heat source. A fraction of the received heat is stored within the given mass while the remaining heat is
Figure 2.7: Time history of the PCM temperature in non-dimensional form at various vertical locations in the PCM domain for four heat fluxes: (a) 6070 W/m², (b) 8150 W/m², (c) 10,030 W/m² and (d) 11,475 W/m². $T_m$=PCM melting temperature, $T_s$=Average surface temperature of the heat source and $T$=instantaneous temperature of PCM domain.
transferred to the adjacent PCM mass on the other end (i.e. away from the heat source). By storing the heat, the PCM mass undergoes the phase change process. Once reached the fully liquid state, it becomes part of the melted liquid domain underneath and the convective motion induced in the liquid domain extends into this newly melted region. The convective cells formed inside the liquid domain effectively transfer heat from the high temperature lower end adjacent to the heat source, to the low temperature upper end at the solid-liquid interface. This not only enhances the heat transfer rate but also induce effective mixing within the liquid domain hence, harmonizing the liquid PCM temperature. Once reached this quasi-steady state, the main role of the liquid PCM mass is to simply transfer the heat that it receives from the higher energy end to the lower energy end. Hence, the temperature rises of a given solid PCM mass increase in a nonlinear fashion due to a continuous increase in the heat transfer rate across the solid-liquid interface as the liquid domain expands. That is, at a given location within the solid PCM domain, the temperature rises at a slower rate when the solid-liquid interface is farther away from that location and a gradual increase in the rate of temperature rise occurs as the interface is approached and a very sharp temperature rise occurs when the interface physically interacts with the solid mass at this location. The shape of the temperature profiles in figure 2.7 confirms this physical description. The results also show that the steady liquid temperature in the region very close to the heat source is slightly higher than that in the bulk liquid domain. This could likely be due to the additional heat transfer into the liquid mass in the close vicinity of the heat source via thermal radiation due to the high surface temperature of the tube.

Similar temperature trends have been reported in the literature during the melting phase of various types of PCMs both experimentally and numerically [10, 44-47]. Chang and Groulx [10] reported analogous temperature trends from their experimental work using dodecanoic acid as a PCM. They argued that the increased heat transfer caused by the larger temperature difference promotes faster melting at the beginning of the heating process, which in turn leads to a faster onset of natural convection, resulting in rapid melting for the rest of the PCM. Adriano Sciacovelli et al. [44] also observed a bump in the temperature profile similar to the present results, which they attribute to the passage of the melting front through the local thermocouple. Anica Trp [45] found similar temperature behavior and
argued that during the phase transition both conduction and natural convection in liquid phase occurred and as melting proceeded, natural convection is enhanced.
Figure 2.8: PCM temperature in non-dimensional form at various heat fluxes versus the normalized time at four different heights from the heat source. (a) 2.5 mm, (b) 10 mm, (c) 35 mm and (d) 55 mm. $T_m$ = PCM melting temperature, $t_s$ = time when the PCM temperature became steady at the given location.
Figure 2.7 shows the rate of temperature change in the PCM domain at different locations at a given heat flux. To further investigate how the heat flux influenced the rate of temperature change, the temperature profiles at a given spatial location were compared for different heat fluxes in figures 2.8(a-d). Note that the time is normalized by the time when the given temperature profile becomes quasi-steady ($t_s$). The plots show that the rate of temperature change is a function of both spatial location and the heat flux. Near the heat source, the temperature rise starts at smaller values of $t/t_s$, and gradually reaches to $t/t_s = 1$. The effect of heat flux is more dominant in the early state of heat transfer. At lower heat fluxes, the temperature rise is gradual from the beginning of the heating process until $t/t_s = 1$. As the heat flux increases, the temperature remains relatively uniform in the early stages of heating and then sharply rises to the quasi-steady state. The plots also show that an increase in the distance from the heat source, extends the time range of the initial near-uniform temperatures at high heat fluxes. For example, at heat fluxes of 10030 and 11475 W/m², the near-uniform temperature range extends to approximately $t/t_s = 0.6$ and $0.8$, as the distance from the heat sources increases to 35 and 55 mm, respectively. It is also observed that the rate of temperature rise becomes almost independent of the heat flux at higher heat flux magnitudes away from the heat source.

2.4.4 Effects of Fourier number ($Fo$) on the interface behavior

The temperature data was used to estimate the Fourier number ($Fo$) which is a dimensionless parameter characterizing the transient heat transfer behavior as a ratio between thermal diffusivity and thermal energy storage rate [48]. Figure 2.9 (a) shows the Fourier number versus the height of the liquid domain at various heat flux conditions. The plot shows that the Fourier number initially increased with an increase in the liquid domain and then gradually became uniform at later stages of melting when the liquid fraction became greater than 70%. This indicates that in the early stage of melting, the heat diffusion is dominant relative to the heat storage. However, as the liquid fraction becomes significant, the diffusion is balanced by the energy storage. The plot also shows that the Fourier number is higher at lower heat fluxes. This implies that the heat diffusion is dominant over a longer period of time compared to that at higher heat fluxes. The time required for the melting of the entire PCM domain is an important parameter in the
designing of a thermal storage. The Fourier number corresponding to the time when the PCM was completely melted ($Fo_m$) could be used to characterize the melting time in a non-dimensional form.

Rayleigh number ($Ra$) is a non-dimensional parameter that characterizes the magnitude of buoyancy-driven force to viscous dissipative forces in the fluid motion [49]. It was calculated using the following equation [50]:

$$Ra = \frac{g\beta(T_S - T_B)L^3}{\nu\alpha}$$

Where, $g$ is the acceleration due to gravity, $\beta$ is the coefficient of thermal expansion of the liquid PCM, $T_S$ is the surface temperature of the HTF tube, $T_B$ is the bulk temperature liquid PCM domain, $L$ is the characteristic length (liquid domain height from the heat source), $\nu$ is the kinematic viscosity and $\alpha$ is the thermal diffusivity of the liquid PCM.

Figure 2.9(b) plots $Fo_m$ versus the $Ra_m$ (i.e. the Rayleigh number when melting completed). The results show as expected, that the non-dimensional melting time decreases with an increase in the buoyancy effects due to an increase in the input heat flux. The results also show that the relationship is nonlinear. Christiano and Luiz [11] also observed that the melting time decreases nonlinearly with an increase in the Rayleigh number.

Figure 2.9(c) shows the variation in the Fourier number ($Fo_m$) versus the Stefan number ($Ste_m$), both at complete melting. The results show that the Fourier number decreases nonlinearly with an increase in the Stefan number. This implies that the melting rate is much faster when the heat storage in the PCM medium has higher sensible heat fraction. This result indicates that a PCM with higher latent heat fraction melts slowly. Christiano and Luiz [11] also observed a decrease in the melting time with an increase in the Stefan number in a nonlinear manner. Figure 10 shows the variation in Stefan number versus the input heat flux. The results show an almost linear increase in the Stefan number with an increase in the heat flux. This indicates that as the input heat flux increases, the relative contribution of sensible heat transfer increases. This is due to the reason that an increase in the heat flux, increases the heat source and melting temperature difference relative to the enthalpy of phase change and hence, the sensible heat transfer increases.
Figure 2.9: (a) Fourier number ($F_o$) versus the normalized height of the melted domain at different heat fluxes. (i) 6070 W/m$^2$, (ii) 8150 W/m$^2$, (iii) 10,030 W/m$^2$ and (iv) 11,475 W/m$^2$. $H$ is the height of the PCM domain above the heat source.
Figure 2.9: (b) Fourier number at complete melting ($F_{om}$) versus the corresponding Rayleigh number ($R_{am}$) for different heat fluxes. (c) $F_{om}$ versus Stefan number at complete melting ($Ste_m$) for different heat fluxes

Figure 2.10: Stefan number ($Ste$) versus heat flux ($q''$) at different input heat flux
2.4.5 Effects of melt fraction on convective heat transfer coefficient, $h$

The convective heat transfer coefficient is calculated using Newton’s law of cooling [50]:

$$ q'' = h(T_{Surf} - T_{Bulk}) $$

Where $q''$ is the input heat flux supplied by the HTF fluid, $A$ is the external surface area of the metal tube in contact with the PCM, $T_{Surf}$ is the average surface temperature of the heat source and $T_{Bulk}$ is the average temperature of the liquid PCM. Since the heat transfer surface area was constant and the average surface temperature showed nearly a constant value, the convective heat transfer coefficient for a particular input heat flux is primarily dependent on the average temperature of the liquid domain. Since, the liquid fraction of the PCM changed continuously throughout the melting process, it caused changes in $T_{Bulk}$, which resulted in the variation in the heat transfer coefficient during the melting process. The knowledge of the temperature distribution within the fluid domain is used as the basis for determining the heat transfer coefficient in this transient process. The heat transfer coefficient at different stages of melting was computed in a transient way with the progression of melting. That is, the transient bulk temperature was estimated as the average temperature of the PCM liquid domain at a given instance. The schematic of the storage chamber in figure 2.2(a) and (b) shows the thermocouples arrangement inside the PCM domain in a set pattern. The transient bulk temperature was estimated at times when a given row of thermocouples was fully occupied by the liquid phase. At that instance, the bulk temperature was estimated as the mean from all thermocouples that were resided in the liquid domain. In the region near the heat source, there was only one column of thermocouples in the middle section of the chamber and hence, the data from those thermocouples were used in the computation of the bulk temperature. The process continued until the PCM completely melted until the top row of the thermocouples. Hence, eight values of the local heat transfer coefficient were computed as a function of the melted fraction.

Figure 2.11 shows the convective heat transfer coefficient ($h$) versus the melted PCM fraction for four heat flux cases. At lower values of the melted fraction i.e. the early stages
of melting, a sharp rise in the $h$ values is observed near the heat source, which then became gradual with a further increase in the melt fraction. This trend was consistent at all four heat flux cases. The results show that the sharp rise in $h$ occurs when the liquid fraction is less than 0.2. It is also observed that the rise in $h$ beyond that fraction has a dependency on the input heat flux. At the lowest heat flux, the value of $h$ remained almost constant during the rest of the melting process. As the input heat flux magnitude increased, a gradual increase in the $h$ value with an increase in the melted fraction is observed. A plausible explanation for the observed trends is that at early stages of melting, the local volume of the PCM domain in the vicinity of the heat source quickly reached the melting temperature and a liquid domain with very strong convective current is developed. This fast and strong convective motion resulted in quick heat transfer through the newly formed liquid PCM domain resulting in a higher rate of heat transfer at the early stage as compared to that at later stage of the melting process when the solid-liquid interface is comparatively far away from the heat source having a lower thermal potential between them.

![Figure 2.11: Convective heat transfer coefficient ($h$) as a function of the melted fraction at different heat fluxes](image)

During the melting process, liquid fraction continued to increase until the PCM completely melted. The average temperature of the liquid PCM is higher in the early stages of melting
since, the melted PCM domain was in the vicinity of the heat source. As the melting continued, the liquid domain expanded opposite to the heat source and the average temperature of the liquid domain first decreased and then remained almost constant, when the liquid fraction became dominant relative to the solid fraction. This nearly constant liquid domain temperature in the latter part of melting is also attributed to the convective cells that became fully established once the liquid fraction became dominant and hence, they regulated the transport of heat. This caused a near-steady value of the heat transfer coefficient when the liquid fraction became dominant.

2.4.6 Heat transport scaling

Natural convection represents a regime where the flow is driven by the buoyancy forces within the fluid. The correlation between the Nusselt number ($\text{Nu}$) and Rayleigh number ($\text{Ra}$) is an important constituent in understanding the natural convective heat transfer process. In the present geometry where the liquid fraction and its bulk temperature continued to change during the melting process, both Nusselt and Rayleigh numbers showed transient trends. The Nusselt number, which is a ratio between the convective to conductive heat transfer, was computed as [48]:

$$\text{Nu} = \frac{h l}{k}$$

(4)

Where, $h$ is the local convective heat transfer coefficient, $l$ is the corresponding local characteristic length (in the present case, it is considered as the liquid domain height from the heat source) and $k$ is the thermal conductivity of the liquid PCM.
Figure 2.12: Nusselt number \((Nu)\) versus Rayleigh number \((Ra)\) at different heat flux conditions. The empirical correlations from Churchill-Chu** [51] and Lim-Bejan* [52] are plotted for comparison.

Figure 2.12 plots the transient Nusselt \((Nu)\) number versus the Rayleigh \((Ra)\) number for four heat fluxes. It is observed that for each heat flux, the Nusselt number gradually increases with the Rayleigh number i.e. with the growth of the liquid PCM domain. The results also show that the data for all heat fluxes followed a consistent trend. The trend line was obtained by using the power law fit and is presented as,

\[
Nu = 0.0781Ra^{0.359}, \quad 1.5 \times 10^3 < Ra < 9.7 \times 10^8
\]  

(5)

To the best of authors’ knowledge, we could not find in the literature a \(Nu-Ra\) relationship during the melting phase with a bottom heat source. Hence, \(Nu-Ra\) relationships for the closest possible conditions were plotted for a cautious comparison. Churchill and Chu [51] presented \(Nu-Ra\) correlation for single phase natural convection above a horizontal heated cylinder. Their correlation is expressed as:
\[ Nu = \frac{hl}{k} = \left\{ 0.60 + \frac{0.387Ra^6}{8} \left[ 1 + \left( \frac{0.559}{Pr} \right)^{\frac{9}{16}} \right] \right\}^2, \quad 10^{-5} < Ra < 10^{12} \& 0 < Pr < \infty \] (6)

Where, \( Pr \) is the Prandtl number. In comparison with the present dataset, it is observed that the trends are relatively similar but at a given Rayleigh number, the Nusselt number values predicted by Churchill and Chu [51] correlation are higher than that observed in the present case. This difference could be due to the reason that Churchill and Chu correlation considers single phase natural convection where the liquid domain size is fixed and the convective cell were well established, while in the present case the liquid domain size continuously increased during the natural convection process and the convective cells undergo developmental stages.

Lim and Bejan [52] presented a correlation for the melting phase of a PCM when the domain was heated from a sidewall. Their correlation is expressed as:

\[ Nu = \frac{0.35 \, Ra^{\frac{1}{4}}}{\left[ 1 + (0.143/Pr)^{\frac{9}{16}} \right]^{\frac{4}{9}}} \] (7)

Comparison with the present results show that Lim and Bejan’s [52] correlation predicts lower Nusselt number values at a given Rayleigh number. This is an expected trend since, in their setup, the melting progression is primarily due to heat diffusion from the side and the buoyancy forces and convective cells formed in the melted fraction did not contribute significantly to the heat transport towards the solid phase.
2.5 Discussions

The melting of a PCM was investigated in a transient manner inside a two-dimensional chamber with a bottom heat source. The solid-liquid interface data and the PCM temperature fields provided a good insight into the spatio-temporal behavior of the PCM during melting and the associated heat transfer. The results show that in the initial stages, the temperature of the PCM in the vicinity of the HTF tube increases through conduction and hence, the PCM temperature quickly passes the melting temperature soon after the heating starts. The phase change starts once the melting point is reached, forming a thin layer of liquid PCM around the heat source. The heat transferred into the remaining solid-phase PCM above the melted domain continued through conduction. In the early stages of melting, the solid fraction is dominant and hence the heat transfer into the PCM domain is primarily through conduction. As the melting progresses, the liquid fraction continues to grow and the convection heat transfer becomes dominant. As one fraction increases and the other decreases, the relative thermal resistance also changes. This effect is manifested in the variation in the heat transfer coefficient with the change in the liquid fraction. Since the heat input from the source remains constant, the variations in thermal resistances influence the local PCM temperature as observed in figures 2.4 and 2.7. As expected, the PCM domain temperature is highest at a location closest to the heat source, which gradually decreases towards the top end (figures 2.4 and 2.7).

The temperature of the HTF decreases along the tube as expected, introducing spatial non-uniformity in the input heat transfer rate, and hence the temperature of the liquid PCM near the inlet end of the chamber is higher, which gradually decreases towards the outlet end. This temperature signature in the PCM is prominent during the early stages of melting. As melting progresses, the temperature in the bulk melted PCM domain tends to become more uniform both axially and vertically.

The results show that the interface profile is relatively linear during the early stages of melting, which later becomes nonlinear. A plausible explanation of this trend is that at early stages of melting, multiple smaller size convective cells are formed within the liquid domain in the vicinity of the heat source that are equally distributed along the domain.
length. Hence, their effect on the heat transfer rate is relatively uniform. Once the liquid domain grows bigger, these convective cells grow and merge to form few larger convective cells. These convective cells increase mixing and hence the local heat transfer enhancement that causes a faster movement of the solid-liquid interface in the vicinity of the convective cell relative to the regions away from the cell inducing nonlinearity in the interface movement. These nonlinear interface profiles hence provide an estimation of the size of these convective cells as seen in figure 2.5. The results in figure 2.11 show that the heat transfer coefficient increases rapidly in the early stages of melting and then it becomes relatively constant. This could be due to the changes in the structure of the convective cells. A plausible explanation for this trend is as follows. In the early stages of melting, the convective cells grow as the height of the liquid domain increases, which contributes to an increase in mixing and hence the heat transfer coefficient increased rapidly. As the liquid domain continued to grow the convective cells become more stable and start to merge. Once the melted fraction exceeded half of the domain size, large convective cells emerge that stabilize the mixing pattern and hence the heat transfer coefficient becomes more uniform. This explanation is further confirmed by the variation in the Fourier number at a given heat flux (figure 2.9a). As the figure shows, the Fourier number increases rapidly in the early stages of melting indicating that the heat diffusion rate is increasing relative to the liquid domain size, which is likely contributed by the convective cells. With a further increase in the liquid fraction, the Fourier number becomes almost constant implying that the rate of heat diffusion is balanced by the expansion of the liquid fraction in the storage domain.
2.6 Conclusions

An experimental study was conducted to investigate the transient heat transfer process during melting phase of a PCM inside a rectangular chamber heated from below. Multiple temperature measurements through thermocouples were used to characterize the thermal behavior of the PCM and an imaging technique was used to capture the movement of the solid-liquid interface. Four different input heat fluxes of 6030 W/m², 8150 W/m², 10030 W/m² and 11,475 W/m² were considered during the experiments. Results show that the solid-liquid interface transformed from a near-linear to the nonlinear form as the melted fraction increased. This trend is attributed to the dominant convective cells that locally enhance the melting rate. Results also show that the magnitude of the solid-liquid interface velocity increased with an increase in the input heat flux, as expected. The results also show that the heat transfer rate significantly affected the melted (liquid) phase likely due to the presence of the convective motions. The simultaneously recorded temperature data and the images were analyzed and that enabled to characterize various important thermophysical parameters associated with the PCM during the melting process. The results show that the convective heat transfer coefficient increased rapidly in the early stage of melting and became nearly steady after approximately two-third of the PCM was melted. This is attributed to the evolution and growth of the convective cells in the liquid PCM domain. A new correlation for the Nusselt number and Rayleigh number is proposed to characterize the heat transfer during the melting phase with a bottom heat source.

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Chapter 3

3 Natural convection melting process in a PCM based thermal energy storage with a U-shaped heat source

3.1 Abstract

Phase change material (PCM) based thermal energy storage system (TES) plays central role in various engineering applications. The geometry, surface and the orientation of the heat source influence the heat transfer behavior during solid-liquid phase transition. By varying inlet conditions of the heat transfer fluid, the study was conducted at four heat flux conditions in a rectangular chamber with a cylindrical U-shaped heat source imbedded in the PCM to investigate the changes in various thermo-physical properties during the phase transition period in a transient manner. The results showed that the heat transfer behavior, interface movement and the heat transfer coefficients differed both axially and vertically inside the chamber as well as with the melting rate. The local convective heat transfer coefficient, \( h_{\text{local}} \) in the inner region, enclosed by the U-tube, was observed 35% higher than the outer region when the heat flux was increased from 3450 W/m\(^2\) to 5840 W/m\(^2\). Stronger convective flow and a lower viscosity owing to higher temperature in the inner region is believed to have caused faster melting than outer region. When the melted domain grew vertically from 15% to 100%, the overall \( h \) in the inner region increased by 40-55% for the lowest to the highest heat flux respectively. The melting rate was also found comparatively higher up until 65-70% of total PCM volume melted before the rate declined because of the higher contributions of the melt fraction from inner region. It was also observed that the Rayleigh number increased approximately 23% in the inner zone and 18% for the whole domain as the heat flux increased from 3450 W/m\(^2\) to 5840 W/m\(^2\).

Keywords

PCM, U-shaped tube, latent heat, thermal energy storage, heat transfer coefficient, interface, melt fraction
3.2 Introduction

Thermal Energy Storage (TES) is a mean to store thermal energy) for a certain period of time for its intended later use [1]. In general, thermal energy storage systems improve the energy efficiency by reusing waste heat as well as by reducing the mismatch between supply and demand [2]. TES is very crucial component in an energy system that is associated, especially with the renewable energy sources since these sources are intermittent by nature and can fluctuate unpredictably, such as solar energy [3]. TES also improves the performance and reliability of an energy system by plummeting peak operating loads and help the system to work within an optimal range [4]. Hence, TES has the enormous potential to perform numerous roles in conserving energy which in turn help to attain energy sustainability [5, 6, 7].

To store thermal energy in a TES, a physical media is needed which would hold the heat. Heat can be stored in two ways, either using a physical method or by applying thermo-chemical energy method. Using physical techniques, heat can be stored in a sensible or latent heat form [8]. In sensible heat storage, the thermal energy is stored by simply increasing the temperature of the storage media, although the mass and the material property play important roles. On the other hand, latent heat storage is based on the capture/release of energy when a material undergoes a phase change typically between solid and liquid phases. Latent heat thermal energy storage (LHTES) is particularly attractive because of its two important features, firstly, it can provide a high-energy storage density (amount of heat per unit mass) and secondly, it has the capacity to store energy nearly at a constant temperature or over a small range of temperature variations around the phase transition temperature [9, 10].

The LHTES are by far the most potential and proven solutions to realistically and effectively offset the drawbacks to utilize renewable energy resources [2, 11-15] and have been used in a diverse range of industrial applications, such as metal processing, food preservation, solidification of castings, building systems, and electronics cooling [8, 16-20]. However, for their effective use as potential heat storage media, desirable thermodynamic, kinetic and chemical properties of these phase-change materials (PCMs)
must first be carefully assessed [21]. Furthermore, longevity, economic and the environment effects of these PCMs must also be taken into considerations [3, 22].

The design of the heat source that carries the heat transfer fluid (HTF) is one of the key factors that dictates the overall heat transfer rate and performance of the storage system. During heat addition/extraction to/from the PCM, the heat transfer occurs in both solid and liquid domains of the PCM. Since the fraction of each phase continuously changes (one increasing while, other decreasing) the local heat transfer within the PCM domain is transient in nature. However, the correlation between the average heat transfer and the melting rate has not been well established [23]. Conduction is the heat transfer mode in the solid phase while, natural convection governs the heat transfer in the liquid phase. The rate of heat transfer is typically higher in the liquid domain compared to solid due to the presence of convective motion induced by buoyancy, which accelerates the transportation of heat. As the buoyancy acts vertically (aligned with gravity), the orientation of the heat source/sink plays a crucial role in the generation of buoyancy-induced flow and its magnitude and extent. This in turn, directly influences the heat transfer within the PCM domain. Hence, an improved understanding of the natural convection process within the liquid domain with respect to the heat source/sink orientation is vital to design efficient thermal energy storage.

Gau and Viskanta [24] reported the presence of convective flow patterns through the visualization of the melting process of paraffin wax in a rectangular enclosure heated from below. Diaz and Viskanta [25] observed that multiple Benard-like cells are formed during initial stages of the melting n-octadecane through a bottom heat source, which grew and merged into a single cell with time. The growth of buoyancy-driven instabilities that controls the convection heat transfer, along with the upward movement of the solid-liquid interface increases the complexity of the problem [26]. However, Hsu et al. [27] argued that the time-scale of interface movement is much lower than that of the convective motion and hence its influence on the convective flow can be ignored.

There are limited studies investigating the heat transfer during phase change and most of them considered side wall as the heat source. In one of the pioneering experimental
investigations on the PCM (n-octadecane) melting inside a rectangular enclosure through a heated sidewall, Hale and Viskanta [28] reported the significant effects of convection on the melting pattern, nature of the solid-liquid interface and heat transfer coefficients. In a similar study Ho and Viskanta [29] experimentally and numerically investigated the melting of wax inside a similar chamber. They reported that except for the initial stages of heat conduction period, natural convection significantly affected the melting and was the dominant mechanism and controlled the melting rate and the solid-liquid interface profile. Gadgil and Gobin [30] argued that, in such case, once the PCM started melting, the effects of conduction may be ignored in further heat transfer analysis.

In an experimental investigation during melting and solidification processes of pure gallium inside a rectangular chamber heated from a side wall, Gau and Viskanta [24] correlated the changing volume of the melt with the heat transfer rate through some relevant dimensionless parameters, like the Nusselt number (\(Nu\)) and the Rayleigh number (\(Ra\)). In another numerically study for melting of Tin in a 2D rectangular enclosure heated through a side wall, Christiano and Luiz [31] observed that the total melting time reduced with an increase in the Rayleigh number (\(Ra\)), and argued that this increase (\(Ra\)) is the consequence of the enhancement in the natural convection.

There is a wide variation in PCM-based thermal energy storage system from the design point of view. These includes but not limited to various shapes like flat plate, annular cylindrical or square tubes, micro/macro encapsulations with different shapes of the containers and slurries containing PCM as well. Furthermore, these storage systems may have different arrangements of the heat sources based on practical applications. In addition, the heat source configuration could be a combination of different geometric arrangements which further complicates the problem.

A major challenge in designing a PCM-based thermal storage is the effective heat transfer to and from the PCM medium during the phase change, which greatly influences system efficiency [3]. This heat transfer mechanism is directly affected by the geometric configuration of the heat source/sink. The complexity further arises due to the simultaneous changes in liquid and solid fractions. The geometric configuration of the heat source/sink
in thermal storage system is likely to be more complicated than the simple heat source configurations of side or bottom wall reported in the literature. Hence, there is a need to extend this research to consider more complicated configurations of the heat source/sink that are of practical interest and conduct a thorough investigation of the phase transformation patterns and the associated heat transfer.

The present study is comprised of experimental investigation to improve understanding of the underlying processes during solid-liquid phase transition in a U-shaped configuration. The specific focus is on the investigation of the transient solid-liquid interface movement and the associated convection heat transfer in the melted region.

3.3 Experimental setup

3.3.1 Thermal Energy Storage Chamber

Experiments were conducted in a thin rectangular chamber that served as the PCM storage. A schematic and a photograph of the experimental set up are shown in figure 3.1 (a) and 1 (b). The PCM chamber has the inside dimensions of 100 mm horizontally (L), 100 mm vertically (H) and a depth of 12 mm (W). An optically clear cast acrylic plate (8 mm thick) cut into a U-shaped structure formed the shape the cavity (Figure 3.2a). All the three sides of this U-shaped plate were 25 mm wide. To form the front and the back wall of the chamber, one 5 mm thick plain clear glass pane and an acrylic sheet of same thickness with the same cross-section of 150 mm × 125 mm were used, respectively. Both the front and back sides of the U-shaped acrylic plate were separated from the front glass pane wall and the back acrylic wall respectively by two pieces of 2 mm thick rubber gaskets, cut into the same size and contour of the U-shaped acrylic plate (Fig. 3.2a). The rubber gaskets, which were placed all along the contact area between the main chamber and both the front and back walls were helpful in preventing the fluid leakage and minimizing the heat loss from the chamber to the surroundings.
Figure 3.1 (a): Schematic diagram of the experimental setup

Figure 3.1 (b): Photograph of the experimental setup. The U-tube heat source is shown inside the PCM chamber
Figure 3.2: Schematic of the PCM chamber, (a) Isometric view, (b) Schematic diagram, Front wall- 5 mm plain glass sheet (150mm x125mm), Main chamber- U-shaped 8 mm thick acrylic plate, back wall-5 mm thick acrylic sheet (150mm x125mm), two 2 mm thick U-shaped rubber gasket; locations of the thermocouples are marked with open circles.
A U-shaped tube made of thin walled smooth copper tube (Copper Alloy 122, wall thickness 0.355 mm, outer diameter 4.76 mm) was used as the heat source for the storage chamber. This copper tube, hereinafter referred to as the heat transfer fluid (HTF) tube, since the heat source was a high temperature fluid passing through this tube. The tube was inserted in the chamber from the open top end and was positioned vertically (figure 3.2a and 3.2b). A clearance of approximately 3.5 mm between the tube and the front and the back walls were kept throughout the experiments.

The distance between the two legs (left and right side) of the U-shaped copper tube was approximately 50 mm for most of its vertical length except near the U-bend in the lower region of the chamber. The outer edge of each leg was approximately 22 mm away from the corresponding side wall of the chamber (figure 3.2b). The maximum immersion depth of the tube inside PCM was approximately 95 mm. The total linear length of the tube in contact with the PCM inside the chamber was approximately 250 mm. All sides of the chamber except the front glass wall and the open top end were insulated (FC-2300-LD, Fibre Cast Inc.) to avoid heat losses to surroundings. The storage chamber was vertically attached to a metal frame using bolt and nuts. The upper part of the chamber was left open to room atmosphere to accommodate thermal expansion of the PCM during melting. The open space at the ceiling of the chamber helped minimize the pressure buildup due to the volumetric expansion of the PCM upon melting [32]. The heat transferred through the open top prevented the heat buildup in the chamber and hence simulated the condition of a continuous heat transfer in a real thermal storage as the heat continued to transfer from the heat source towards the boundaries of the storage domain.

3.3.2 Thermal energy storage media

A mixture of fine paraffin wax and copolymer alloys in the form of milk-white pellets commercially available as ‘Polyfin’, was used as the phase change material (PCM) in this study. Various thermo-physical properties of the PCM as per published literature are listed in Table 1. One of the key features of ‘Polyfin’ for its suitability for present research is that it becomes transparent once melted, which was a very useful and crucial feature for
accurately imaging the melting process and tracking the solid-liquid interface. A very small amount of silver coated hollow glass sphere having an average diameter of 9-13µm was used along with the PCM as the tracer particles to investigate the behavior of the molten liquid flow through imaging. The volume change of the wax in the liquid stage was estimated to be approximately 10% from the solid state.

Table 3.1: Thermo-physical properties of PCM at 25 °C (VWR CA 27900268) [33, 34]

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melting Point, °C</td>
<td>55</td>
</tr>
<tr>
<td>Specific Gravity, kg/kg</td>
<td>0.92</td>
</tr>
<tr>
<td>Specific Heat, kJ/kg/K</td>
<td>2.890</td>
</tr>
<tr>
<td>Thermal Conductivity, W/m/K</td>
<td>Solid ~0.24</td>
</tr>
<tr>
<td>Latent Heat of Fusion, kJ/kg</td>
<td>Liquid ~0.18</td>
</tr>
<tr>
<td>Kinematic Viscosity, m²/s</td>
<td>5.50×10⁻⁶</td>
</tr>
<tr>
<td>Thermal Diffusivity, m²/s</td>
<td>180</td>
</tr>
<tr>
<td></td>
<td>7.985×10⁻⁸</td>
</tr>
</tbody>
</table>

3.3.3 Heat source

As mentioned earlier, a U-shaped copper tube acted as the heat source in this study. It was considered as the heat source in this study since the hot air (HTF fluid) was passing through this tube. The HTF continuously passed through the tube at a controlled flow rate and temperature. In order to minimize the pressure fluctuations in the HTF, the incoming compressed air from the laboratory supply line was initially passed through a settling tank to dampen pressure fluctuations. Downstream of the air settling tank, a pressure regulator was employed to control the pressure of the air in the system. The air was then passed through a flow meter to control its flow rate before passing through the heater (figure 3.1). The controlled air was then heated while passing through an air heater (Zesta ZA0017) equipped with a PID controller (Zesta ZEL-9100) to precisely control the temperature of the HTF fluid. The accuracy of the PID controller was within ±2°C. The air flow rate was maintained at 40 liters/min. The accuracy of the flow meter was estimated as ±5%. Four different controlled temperatures of the HTF were used as inlet temperatures (150 °C, 175 °C, 200 °C and 225 °C) by adjusting the heater controller, which provided four different input heat fluxes of 3450 W/m², 4370 W/m², 5020 W/m² and 5840 W/m², respectively.
3.3.4 Temperature data

A total of 28 T-type thermocouples (T30-2-507, Zesta Engineering Ltd.), hereinafter referred to as TC were used to measure the temperatures at various locations in the experimental setup, shown in figure 3.2(b) as small black open circles. Out of these 28 thermocouples, two thermocouples were placed in direct contact with the air flow to measure its (HTF) temperatures at the inlet and outlet of the PCM chamber. The temperature on the surface of the HTF tube inside the chamber was measured at eleven different locations along the tube length. Out of these eleven thermocouples, ten thermocouples were affixed on the surface of the two vertical legs (five TCs on each leg)) with an equal spacing of about 20 mm between them, and the other thermocouple was fixed in the middle of the curved part of the U-shaped tube (figure 3.1b). All these eleven thermocouples were affixed at the back side of the tube to avoid any imaging obstructions. For measuring the temperature inside the PCM domain, 15 thermocouples were arranged in an array format (5 rows and 3 columns), with a column on left, right and middle of the U-tube (figure 3.2a and 3.2b). The left and right columns were located 14 mm from their respective side walls. The center thermocouples’ column was located at the vertical center line of the chamber and tube, at a distance of 50 mm from each side wall. The vertical distance between each thermocouple row was also kept at 20 mm and they were at the same vertical height as that of the tube surface thermocouples (figure 3.2a and 3.2b). The bottom thermocouples’ row was located approximately 5 mm above the curved surface of the HTF tube near the bottom of the chamber. All thermocouples in the PCM domain were inserted individually into the chamber through the insulated acrylic back wall.

The temperature data from all 28 thermocouples were recorded simultaneously at a rate of 3 Hz via data acquisition cards (thermocouple modules: NI-9214 and NI-9211) controlled by the LabVIEW software. To ensure accuracy and similarity, the temperature data recording for all experimental cases were started 1 min prior to the beginning of the heating process and continued 5 min after the PCM was completely melted.

3.3.5 Calibration of thermocouples
Calibration of all thermocouples was accomplished by using an insulated hot water bath. A high accuracy thermometer (Kessler 10/30 ST Jointed Thermometer) was used as the reference. The calibration was conducted over a temperature range from 90 °C to 0 °C with increments of about 10 °C between each reading. All calibration curves showed almost linear relationship. The uncertainty of temperature measurements from thermocouples was estimated as ±0.4%.

### 3.3.6 PCM chamber Insulation

As mentioned earlier, the front wall was made of a clear glass pane. This was required for this study since the clear glass pane allowed optical access during image acquisition of the melting process inside the chamber by a high resolution camera. Since this was the only un-insulated wall of the chamber, some heat loss occurred through this glass pane. Hence, it was necessary to ensure that this heat loss from the front wall did not influence the melting process inside the chamber. This was verified by the authors in an earlier study where the transient behavior of the PCM temperature field was compared for the storage chamber with the front glass pane and fully insulated storage chamber with the same heat source geometry (horizontal tube). For that study, the range of input fluxes was the same as in the present study. The results show that the PCM temperature within the domain at each heat flux case was slightly higher for fully insulated chamber compared to that with the front glass pane, as expected. The difference in the mean temperatures for the two chamber configurations was on average 8.7%. Because of the transient nature of the process, the temperature profiles at different locations within the PCM domain over the entire melting period were also compared for both cases. The results show no significant variations in the trends for both insulated and un-insulated cases hence, confirming that the un-insulated pane had no considerable influence on the underlying transient melting process.

### 3.3.7 Image acquisition and image processing
A 4 Mega Pixel high resolution charged-coupled device (CCD) camera (Flare 4M180-CL) with a resolution of 2048 × 2048 pixels was used to capture the images of the phase change process and to track the solid-liquid interface and the flow behavior of the melted PCM. The camera was connected to an image acquisition system (DVR express core, IO Industries) to record images. The camera was controlled via the DVR express CoreView software installed on a PC. The field of view of the camera was set at 94 mm × 94 mm. For each run, images were captured from the beginning of the melting process until the PCM was completely melted. The camera frame rate was set at 40 frames per second (40 Hz) for different experimental runs irrespective of the input heat flux. The stored images were transferred from the DVR core to an external storage media in suitable format for further processing.

3.3.8 Experimental data analysis

The heat gain by the PCM was considered equal to the heat loss by the HTF passing through the PCM domain inside the chamber and was calculated from the following equation [35]:

\[
\dot{q}_{\text{total}} = m c_p (T_{\text{in}} - T_{\text{exit}}) \tag{1}
\]

where, \(\dot{q}_{\text{total}}\) is the heat transfer rate, \(m\) is the mass flow rate of air, \(c_p\) is the specific heat of air, and \(T_{\text{in}}\) and \(T_{\text{exit}}\) are the inlet and exit temperatures of the air (HTF). Since the thermocouples to measure inlet and outlet temperatures were located approximately 1 cm off from the physical entrance and exit of the PCM in its full melted condition, the heat loss from the HTF tube between \(T_{\text{in}}\) and \(T_{\text{exit}}\) thermocouple locations and the chamber inlet and outlet locations, respectively, were calculated. The net heat transferred to the PCM (\(q_{\text{net}}\)) was then calculated by subtracting the inlet/outlet heat loss, from the total heat lost by the HTF. The net heat flux (\(q''_{\text{net}}\)) into the PCM domain was computed by dividing (\(q_{\text{net}}\)) by the surface area of the HTF tube inside the domain. The uncertainty in estimating the heat flux was approximately on average within ±10%. (see appendix C).

As the liquid fraction continued to increase with the progression of melting, the local heat transfer coefficients were computed when the melted fraction reached up to a particular
thermocouple. Because of the nature and pattern of melting in a U-shaped tube configuration (heat source), the local heat transfer coefficient \( (h) \) was measured based on the change in the vertical extent of the melted PCM in the vicinity of the heat source. It is the same approach used for natural convection heat transfer from vertical tubes [36]. Hence, the local heat transfer coefficient values were obtained in each zone with the change in the height of the melted PCM. The vertical locations of thermocouples were considered as the reference lengths i.e. the local heat transfer coefficient in each zone was computed when the melted PCM domain reached subsequent thermocouple within a given column in the respective zone. At a given vertical location, the local heat transfer coefficient was computed using the equation,

\[
q'' = h (T_s - T_{bulk})
\]

where, \( q'' \) is the heat flux assumed to be equal to the average heat flux through the HTF tube, \( T_s \) is the local surface temperature of the HTF tube and \( T_{bulk} \) is the local bulk temperature of the melted PCM.

Note that the tube surface temperature and bulk liquid temperature were computed as the average from all corresponding thermocouples that were immersed in the liquid PCM in each zone.

3.3.9 Interface detection

Tracking the solid-liquid interface and detecting its form as the melting progressed, provided an in-depth insight for the better understanding of the underlying process. Using an in-house algorithm in a Matlab platform, the solid-liquid interface was detected in each image over the whole melting process. The algorithm initially preprocessed raw images to increase their signal-to-noise ratio by rescaling in the gray-scale values between solid and liquid phases (upper end gray-scale values correspond to the solid PCM and lower end gray-scale values correspond to the liquid PCM) for accurate identification of each phase and its transformation into a binary format for the precise detection of the interface. A series of morphological operations were performed to remove any noise from the data. The coordinates of the detected interface in the binary image were recorded at every spatial
location which when processed generated the actual profile and position of the solid-liquid interface at a given time. Because of the physical presence of various thermocouples in the PCM domain, the interface profile in the close proximity of a thermocouple was locally influenced. Hence, in the subsequent analyses, the interface data in the locality of each thermocouple were excluded. The uncertainty in detecting the interface was within ±2 pixels that correspond to ±90 µm. Figure 3.3 shows an instantaneous image of the PCM domain. The detected solid-liquid interface in the corresponding image is also plotted with a red line.

![Figure 3.3](image)

**Figure 3.3:** A snapshot showing the real image of the chamber during PCM melting for one experimental run (heat flux of 4370 W/m²), the interface is marked as the red line, the U-tube is visible by two thick vertical white lines, the white area represents the solid phase and the dark-gray area represents the melted PCM.

3.4 Results
The objective of this study was to study the heat transfer phenomena during the phase change process and the influence of the geometry of the heat source. The temperature data acquired throughout the storage chamber, along with the simultaneous imaging provided detailed information about the thermo-fluidic behavior during the solid-liquid phase change process.

### 3.4.1 Colormap of the temperature fields

It was mentioned earlier in the experimental setup section that the temperature was measured at 26 locations inside the chamber (11 locations on the surface of the tube and 15 locations inside the PCM domain) to characterize the thermal behavior of the PCM during the melting phase. The variation of the temperature field inside the chamber at different stages of melting is illustrated in Figure 3.4 (left side) along with the corresponding images showing the melting trends through the front glass panel (right side). Figure 3.4(a) shows the data soon after the beginning of the heating process. At this stage, heat transfer is in the early phase via conduction through the PCM. The heat lost by the HTF is evident through a gradual decrease in the tube surface temperature. The bulk PCM temperature at this stage is not significantly influenced by the heat transfer and the PCM domain remains in the solid phase (see the corresponding image on the right).

As the heat transfer to PCM continued and, once the melting temperature of the PCM was reached, the solid phase of the PCM started to melt. The U-tube surface in contact with the PCM naturally was at the highest temperature near the top left corner of the chamber where the HTF tube entered the PCM domain. Since the HTF was continuously losing heat to the PCM, the temperature of the HTF decreased along the tube, which caused an increase in the PCM domain temperature as well as a decrease in the tube surface temperature. Due to the higher HTF and hence the tube surface temperature, the melting process initiated near the top left corner of the chamber (figure 3.4b). As the time passed, the melted zone started to expand downward and horizontally. Thus, the pool of melted PCM acquired the shape of an inverse cone and continued to grow.
Figure 3.4: (Left) Colormaps of the temperature field in the PCM chamber at different stages of melting. Center area marked by black lines in U-shape profile indicate the location of the HTF tube, (Right) corresponding image of PCM domain.
Following the formation of the melted zone in the entrance region of the HTF, another smaller melted region having a similar conical shape quickly developed around the HTF tube where it exited the PCM domain near the top right half of the chamber (figure 3.4c). This is an unexpected trend because as the HTF loses its temperature along the tube, one expects that the melting front to progress along the same path. A plausible explanation for the observed trend is described as follows. As mentioned above, the melted zone formed in the HTF entrance zone started to expand horizontally and vertically, indicating the movement of the solid-liquid heated interfacial front. The heat transfer through this interfacial front was higher since the convection heat transfer that is induced in the liquid PCM enhances the rate of heat transfer relative to the conduction. Thus, the heat more rapidly transferred horizontally from the left leg of the tube at the upper end of the PCM towards the right leg as well as the left side wall. But since the heat was also being transferred from the right leg near the upper end, the PCM temperature in the vicinity of the right leg reached the melting point faster than the other regions of the PCM. Another interesting behavior observed at this stage was the change in the trend of the tube surface, which corresponds to the similar changes in the HTF temperature. As figure 3.4(a) shows, the tube surface temperature decreased gradually from the inlet to the outlet before the commencement of melting. However, once the melting began and the heat started to propagate more rapidly in the horizontal direction, the surface temperature of the right leg near the outlet became higher than that of the tube on the immediate upstream side almost up to the tube bend at the bottom of the chamber. The reason for this trend is because of the higher PCM temperature in the tube outlet region due to the horizontal heat transfer from the left side, which reduced the temperature difference between the HTF and the PCM causing a rise in the tube surface temperature.

As the melting further progressed with time, the melted fraction continued to grow in the upper region of the PCM chamber surrounding the two tube legs and a new melted zone started to develop just above the U-bend of the HTF tube in the lower mid region of the chamber and the overall temperature of the melted PCM continued to rise. The formation of this third zone about the tube bend is due to the higher contribution of buoyancy. The heat source in this region was near-horizontal, where the buoyancy forces supported the transport of heat to the PCM above the tube i.e. the buoyancy forces aligned with the
direction of the solid-liquid interfacial front. As the melting process further continued, all three melted zones continued to grow (figure 3.4d) and eventually merged (figure 3.4e). A blob of solid PCM remained in the middle of the U-tube which eventually melted. The results also indicate that the temperature gradients are stronger in the melted PCM than that in the solid PCM. It is observed from the image sequence in figure 3.4 that the melting was more rapid in the PCM enclosed by the U-tube (inner zone) that that in the region surrounding the U-tube (outer zone). Once the PCM in the inner zone was fully melted, the melting rate increased in the outer zone. The image sequence also showed that the solid-liquid interface was nonlinear in all three zones and its nature continued to change with the growth of these zones. The interface acquired a relatively linear form once the PCM was fully melted in the inner zone.

3.4.2 Melting rate and melt fraction of PCM

Using an in-house algorithm, the images recorded during the melting process were analyzed to compute the melted fraction in each image. The change in the total melted fraction with time which expresses the instantaneous melting rate is plotted in figure 3.5 for all four heat flux cases. As expected, the melting rate of the PCM is highest for the highest heat flux, which decreased monotonically with a decrease in the heat flux. From the graphs it is observed that at the initial stages for a small period of time, there was no melting at any heat flux. This time duration corresponds to the period during which PCM in the vicinity of the tube reached its melting temperature after the HTF fluid started passing through the PCM domain.

From figure 3.5, it also observed that the melting rate was comparatively higher up until approximately 65-70% of the total PCM volume was melted and after that, the melting rate decreased. A plausible explanation for this trend is described below. As seen earlier in figure 3.4, the PCM in the inner region melted first and at a higher rate due to the presence of two heat sources (i.e. left and right legs of the tube). Thus, the higher melting rate observed in figure 3.5 corresponds to the melting of the PCM in the inner region and the slower rate corresponds to the melting of the PCM in the outer region.
Figure 3.5: Plots showing instantaneous volume of the melted PCM domain as the ratio of the volume of the total domain. Results presented for four input heat fluxes, (a) 3450 W/m², (b) 4370 W/m², (c) 5020 W/m² and (d) 5840 W/m².

3.4.3 Interface profile

As shown in figure 3.3, the solid-liquid interface was detected using an in-house algorithm. The algorithm was used to detect the interface developed in the upper region of the chamber due to the melting caused by the left and right legs of the tube. The purpose was to investigate the melting behavior and the associated heat transfer due to the vertical legs of the heat source. The bottom melting near the U-tube bend was not considered since the interface detection and analysis would have become very complex. The interface was tracked up until the bottom melting zone merges with the top melting zones. Tracking of the interface in each image allowed to analyze the transient behavior of the interface dynamics. Figure 3.6 plots the time history of the detected interface at the heat flux of 4370 W/m² as an example.
The time interval between the successive interface profiles as seen in the plots was 66.67 sec. From the profile dynamics, it is clearly evident that the interface movement was non-linear in both space and time. The interface moved more rapidly near the heat sources as expected. The results also show that the interface movement was not symmetric about a given heat source (left or right leg) i.e. the interface movement was relatively faster on the heat source facing the inner region. Similarly, the interface movement was relatively faster at the left heat source, which was the heat inlet, as expected. Since the HTF has to enter from one leg and exit from other, this behavior is the practical reality due to geometry of the heat source. Hence, for a better characterization of this behavior, the PCM chamber was divided into four separate imaginary zones as shown in figure 3.7, and denoted as zone A, B, C and D. It is to be noted that, the boundary line between zone B and zone C was commonly shared by the column of five thermocouples vertically arranged along the center line of the chamber (figure 3.2). Hence, the same set of temperature data obtained from
these five thermocouples were used for both zone B and zone C to calculate each zone’s parameters. Various thermo-physical parameters were calculated initially by determining the local parameters first and then the average values were taken into account to find the global parameters.

![Diagram of PCM chamber](image)

**Figure 3.7:** Schematic of the PCM chamber showing four separate imaginary zones marked as A, B, C and D by dotted lines

### 3.4.4 Determination of local convective heat transfer coefficient ($h$)

The convective heat transfer coefficient ($h$) in the liquid (melted) PCM was calculated using Newton’s law of cooling [12]:

$$q'' = h(T_{Surf} - T_{Bulk})$$

(3)

Where $q''$ is the heat flux, $T_{Surf}$ is the average surface temperature of the HTF tube that is adjacent to the liquid PCM and $T_{Bulk}$ is the average temperature of the liquid PCM. As
the liquid domain of the PCM expanded with time, the local heat transfer coefficients associated with the top melting zones i.e. formed by the left and right vertical heat sources (zones A, B, C, and D), at different instances were computed using the above equation. The local values of $h$ were computed at time instances when the liquid fraction reached consecutive thermocouples embedded in the PCM domain. Furthermore, similar to the interface detection, the local values of $h$ were computed up until the bottom melting zone merges with the top melting zones. A final value of $h$ was calculated when the inner region was fully melted. It is to be noted that, the moment when the central region bounded by the inner area of the U-tube (zone B and C) was completely melted, still some parts at the lower region in zone A and D were not yet melted completely. Thus, to keep similarity in computations, data from only three upper thermocouples were considered for local $h$ calculations.

The values of local convective heat transfer coefficients ($h_{local}$) were calculated in a transient manner as the melted PCM expanded for all heat flux cases. Figure 3.8(a) and (b) show the values of $h_{local}$ for the lowest and highest heat flux cases (3450 W/m$^2$ and 5840 W/m$^2$) respectively. In the plots, $y$ is measured from the top of the chamber and $H$ is the height of the chamber. Hence, $y/H = 0$, corresponds to the top of the PCM chamber. The results show a general trend that the heat transfer coefficient in the liquid PCM increased with the expansion of the melted domain in each chamber zone (A, B, C and D) on both sides of each HTF tube leg. This increase in the magnitude of $h_{local}$ is due to two main reasons. The first is the increase in the HTF tube contact length with the liquid that strengthens the magnitude buoyancy-induced flow and second is the expansion of the liquid PCM zone which allows the growth of convective cells formed in the liquid zone.
Figure 3.8: Average values of local convective heat transfer coefficient ($h$) at different heights of chamber. Results are presented for two input heat fluxes, (a) 3450 W/m$^2$, (b) 5840 W/m$^2$. 
These convective cells increase the transportation of heat from the source to the other boundaries of the liquid zone. At a given heat flux, the results show a clear influence of given liquid PCM zone relative to the heat source orientation i.e. the \( h_{\text{local}} \) was significantly larger in the liquid PCM zones within the inner region (zones B and C) compared to those in the outer region (zones A and D). It is found that the values of \( h_{\text{local}} \) in the inner region are on average about 35% higher than the values of \( h_{\text{local}} \) in the outer region at both highest and lowest heat fluxes. Relatively larger melted zone volume and higher liquid contact length of the HTF tube in the inner region are the likely causes for higher \( h_{\text{local}} \) values in that region. Comparison of figure 3.8(a) and (b) shows that while the trends of \( h_{\text{local}} \) variations are relatively similar, the magnitude of \( h_{\text{local}} \) increased with an increase in the heat flux.

![Figure 3.9](image)

**Figure 3.9:** Variations in average values of local convective heat transfer coefficient (\( h \)) shown spatio-temporally along the storage chamber. Results presented for four input heat fluxes, (a) 3450 W/m\(^2\), (b) 4370 W/m\(^2\), (c) 5020 W/m\(^2\) and (d) 5840 W/m\(^2\).
Figure 3.8 presented the transient variation in $h_{local}$ in four chamber zones at the lowest and highest flux cases. To further generalize the results, the average value of $h_{local}$ in each chamber zone is presented in figure 3.9 for all heat flux cases. The trends are similar to that observed in figure 3.8, i.e. the highest values of the heat transfer coefficient, $h$, is observed in zone B and then zone C, which are in the inner region. The outer region (zones A and D) have almost similar magnitude. Results also show that in each zone, the value of $h$ increased monotonically with an increase in the heat flux. It is found that the value of $h$ in the inner region (zones B and C) increased approximately by 10% as the heat flux increased from 3450 W/m$^2$ to 5840 W/m$^2$. While in the outer region (zones A and D) the value of $h$ increased approximately by 5% over the same heat flux range.

3.4.5 Variation in local Rayleigh numbers ($Ra$) inside the chamber

Rayleigh number ($Ra$) is a non-dimensional parameter that characterizes the magnitude of buoyancy-driven force to viscous dissipative force in the fluid domain [37]. It is calculated using the following equation [12]:

$$Ra = \frac{g\beta(T_s - T_B)l^3}{\nu\alpha}$$

(4)

Where, $g$ is the acceleration due to gravity, $\beta$ is the coefficient of thermal expansion of the liquid PCM, $T_s$ is the surface temperature of the HTF tube, $T_B$ is the bulk temperature liquid PCM domain, $l$ is the characteristic length (in the present case, it is the liquid contact length of the HTF tube), $\nu$ is the kinematic viscosity and $\alpha$ is the thermal diffusivity of the liquid PCM. The local variations in the Rayleigh number were also calculated in the four chamber zones in the transient manner similar to the computation of $h_{local}$. The average value of the local Rayleigh number in different zones at all heat fluxes are shown in figure 3.10. The results show the highest values of Rayleigh number in zone B, which followed by that in zone C. The values in the outer region (zones A and D) were in general further lower. While overall the trends in Rayleigh number variations are similar to that of the heat transfer coefficient, one prominent difference was in zone C. The values of heat transfer
coefficient in zones B and C were relatively comparable indicating that the flow patterns in both zones are quite similar. However, the Rayleigh number in zone C was significantly lower than that in zone B and approximately in the same range as the Rayleigh number in the outer region. A plausible explanation for this trend is that the Rayleigh number is primarily influenced by the fluid temperature rather than the fluid flow behavior. As seen earlier in figure 3.5, the HTF tube surface temperature was significantly higher in zone B compared to that in zone C and hence the higher liquid PCM temperature in the liquid PCM in zone B, which caused a higher reduction in the liquid viscosity. The results also show that the Rayleigh number increased almost monotonically with the heat flux in the chamber zones. It is observed that the Rayleigh number on average increased by 23.2% in the inner zone and 18.3% for the whole domain as the heat flux increased from 3450 W/m$^2$ to 5840 W/m$^2$.

![Rayleigh number (Ra) at different axial location of the chamber. Results are presented for four input heat fluxes, (a) 3450 W/m$^2$, (b) 4370 W/m$^2$, (c) 5020 W/m$^2$ and (d) 5840 W/m$^2$](image-url)
3.4.6 Determination of the bulk parameters

In the preceding sections, transient behavior of the heat transfer coefficient in different spatial zones is presented. To characterize the overall behavior in a transient manner, the local heat transfer coefficient values spatially averaged over the four zones (A, B, C and D), at a given distance of the propagating melting front (which is moving in a transient manner) are presented in figure 3.11. Considering these facts that the center part was completely melted for all four input heat loads used in this study, the center part of the chamber domain bounded by the two vertical legs of the U-tube was evaluated separately as well. Figure 3.11 shows the $h$ values at various melt fraction which shows a more realistic trend in $h$ values compared to the changes in the two side chambers (A and D) as shown in figure 3.8. Considering this fact, further analysis in this study have been done based on the data from the center part only (zone B and C).

Figure 3.11: Average values of convective heat transfer coefficient ($h$) at different heights of the chamber in the center region of the PCM domain in zone B and zone C. Results are presented for four input heat fluxes, (a) 3450 W/m$^2$, (b) 4370 W/m$^2$, (c) 5020 W/m$^2$ and (d) 5840 W/m$^2$.
The value of the heat transfer coefficient when the inner zone was fully melted are also included. The results show that the local heat transfer coefficient continuously increased as the liquid domain continued to expand and the heat transfer coefficient reached a maximum value when the PCM domain was fully melted. As mentioned earlier, the plausible cause is the stronger buoyancy-induced flow and larger domain for the convective cells to grow. The plot shows a similar trend of the heat transfer coefficient increase at all heat fluxes while, the magnitude of $h$ at a given height increased monotonically with the heat flux. The results show that at the lowest heat flux, the value of $h$ increased by 40% as the vertical distance increased from $y/H =0.15$ to $y/H =1$. Over the same vertical distance range, the value of $h$ increased by 55% at the highest heat flux.

To further characterize the bulk behavior, the heat transfer coefficient values averaged over space and time are plotted in figure 3.12 at all heat flux cases. The results as expected show that the overall heat transfer coefficient increased with an increase in the heat flux, however, the increasing trend is nonlinear. The results also show that the overall heat transfer coefficient increased by approximately 10% as the heat flux increased from 3450 W/m$^2$ to 5840 W/m$^2$ (lowest to highest heat flux).

![Figure 3.12: Convective heat transfer coefficient ($h$) as a function of input heat flux ($q''$)](image-url)
3.4.7 Nusselt number (Nu) correlations

The heat transfer behavior in a given configuration during natural convection is typically generalized in terms of the relation between Nusselt number, $Nu$ (a ratio between the convective to conductive heat transfer,) and Rayleigh number, $Ra$. Such relationship provides an understanding of how the convection heat transfer is influenced by the buoyancy effects. The local Nusselt number was computed as [12]:

$$Nu = \frac{hl}{k}$$  \hspace{1cm} (5)

Where, $h$ is the local convective heat transfer coefficient, $l$ is the corresponding local characteristic length (in the present case, it is considered as the liquid contact length of the HTF tube) and $k$ is the thermal conductivity of the liquid PCM. The computation of the local Rayleigh number has already been discussed earlier. The local Nusselt number is plotted versus the local Rayleigh number in figure 3.13 for all heat flux cases. The results show as expected that the local Nusselt number increased with an increase in the Rayleigh number. To the best of authors’ knowledge, the results in figure 3.13 provide for the first time, the relation between the Nusselt and Rayleigh numbers in the transient melting process using a U-shaped heat source. A generalized expression for this relation is obtained using a power law fit to the experimental data, which is presented as,

$$Nu = 0.0071Ra^{0.4501}, \hspace{1cm} 3.0 \times 10^6 < Ra < 1.5 \times 10^9$$  \hspace{1cm} (6)

The geometry considered in the present study comprised of a combination of vertical (side) heat sources and a bottom heat source. In the literature, a well-established $Nu$-$Ra$ correlation is reported for heat transfer from a vertical cylinder [36, 38, 39] that was originally proposed by LeFevre and Ede in 1956 [40]. The correlation is presented below:

$$Nu_H = \frac{4}{3} \left[ \frac{7Ra_H Pr}{5(20 + 21Pr)} \right]^{\frac{1}{5}} + \frac{4(272 + 315Pr)H}{35(64 + 63Pr)D}$$  \hspace{1cm} [7]

$H$ represents the height of the chamber, $D$ is the diameter of the heat source and the dimensionless Rayleigh and Prandtl numbers are represented by $Ra$ and $Pr$ respectively. It is to be noted that, for this correlation, the medium considered was a single phase fluid.
Lim and Bejan [41] also investigated a two phase system and developed a correlation during the melting phase of a PCM where they considered a heated vertical wall of the chamber as the heat source. Their correlation is expressed as:

\[
Nu = \frac{0.35 \, Ra^{\frac{1}{4}}}{[1 + (0.143/Pr)^{\frac{9}{4}}]^{\frac{9}{16}}}
\]  

(8)

These correlations are plotted in figure 3.13 for comparison. It is observed that the present results agree more closely with the correlation of Lim and Bejan [41] than that of LeFevre and Ede [40]. It is expected since, Lim and Bejan’s [41] correlation is for the phase change process with a vertical heat source while LeFevre and Ede’s [40] correlation is for single phase convection from a vertical heat source. A close comparison of the present results with Lim and Bejan’s [41] correlation show that the Lim and Bejan [41] correlation predicts higher Nusselt number in the lower range of Rayleigh number however as the Rayleigh number increases, the Nusselt number in the present geometry exceeds the Nusselt number predicted by Lim and Bejan’s [41] correlation. This difference in the trend is due to the differences in the geometry of the heat source. The present results in the lower range of Rayleigh number corresponds to the duration when the PCM started to melt from the top and propagated downward and horizontally until it merges with the melted PCM zone emerging from the bottom. The correlation of Lim and Bejan [41] was based on the constant temperature condition of the vertical heat source, while in the present case, the temperature of the HTF tube surface was decreasing due to heat loss from the HTF (figure 3.4). Hence, the buoyancy forces in this case are relatively weaker than that for the case with a constant temperature heat source. This could likely be the reason for the lower Nusselt number values in the current conditions. The comparison also shows that beyond this lower Rayleigh number range, the Nusselt number in the present configuration started to become larger than that from Lim and Bejan [41]. This is likely due to the reason that beyond this Rayleigh number, the bottom melted zone merged with the top melted zones (figure 3.4e), which enhanced the convective currents across the domain due to the additional contribution from the bottom heat source. The Nusselt number values in the higher range of Rayleigh number correspond to the condition when the PCM in the inner region was fully melted (figure 3.4f). In this condition, the buoyancy induced convective
currents from the bottom tube surface supported the flow induced by buoyancy from the vertical sections of the tube and played a significant role in the overall convective motion. Hence, resulted in higher Nusselt number values compared to that predicted by Lim and Bejan’s [47] correlation such heat source was not present in the configuration of Lim and Bejan [47]. Comparison of the present correlation with that of LeFevre and Ede [40] show that at a given Rayleigh number, the Nusselt number values predicted by LeFevre and Ede’s [40] correlation is higher than those in the present study. A plausible explanation for this difference is that LeFevre and Ede’s [40] correlation considers single phase fluid in the entire domain which influenced the convective flow from the initial stage of heating while in the present case, the liquid domain increased with time.

Figure 3.13: Nusselt number (\(Nu\)) versus Rayleigh number (\(Ra\)) at different heat fluxes. Empirical correlations obtained from Lim-Bejan\(^1\) [41] and LeFevre-Ede\(^2\) [40] are also plotted for comparison.
3.4.8 Physical description of the process

Based on the results from the study, the physical explanation of the problem can be described as follows. The phase change material (PCM) stored inside the chamber was heated from its initial solid phase to complete melting stage using a U-shaped heat source imbedded in the PCM domain. The results show that in the initial stages, the temperature of the PCM in the vicinity of the HTF tube increases through conduction and once it passed the melting point, the melting begun. In the early stages of melting, the solid fraction is dominant and hence the heat transfer into the PCM domain is primarily through conduction. Because of the orientation and geometry of the heat source inside the chamber and the direction of the heat flow, the melting point of the PCM at the left top region of the chamber around the vicinity of the HTF tube, where it entered the PCM domain (figure 3.2b) reached first and the melting of the PCM started in this region, as expected. A pool of liquid PCM formed in an inverse conical shape around the tube (heat source) and kept growing with time through continued addition of heat (figure 3.4b). Following the initiation of the melting in this area, melted zones in two more regions, one near the exit of the heat source from the PCM domain near the top right of the chamber and another just above the U-bend region of the heat source, developed as two separate melted entities. Because of effects of virtually two heat sources, formed by the two legs of the U-shaped cylindrical heat source, the inner region melted quickly than the outer regions. The heat transfer behavior, interface movement and the heat transfer coefficient differed both axially and vertically as well as with the melting rate. By varying inlet conditions to four inlet temperature settings for the HTF fluid, the study was conducted in four heat flux scenarios to investigate and better understand the effects of these changes on various thermo-physical properties during the phase transition period in a transient manner.
3.5 Conclusions

An experimental study was conducted for a time dependent heat transfer process where melting of a PCM inside a rectangular chamber was accomplished along with a temperature and an image recording system. The presented results showed that the imposed heat flux and the position of the heat source have a significant effect on the shape/structure and nature of the interface throughout the melting period. The temperature profiles at various location in the PCM domain provided a clear picture of the insights of the heat transfer process. Results showed that the solid-liquid interface moved in a nonlinear fashion and the growth of the interface away from the heat source showed nearly a uniform velocity. Results also show that the velocity of the solid-liquid interface increased with an increase in the input heat flux, as expected. The results also show that the heat transfer rate significantly affected the melted (liquid) phase likely due to the variations in the convective motions. The simultaneously recorded temperature data and the images were analyzed and that enabled to compute various important thermo-physical parameters of the PCM during the melting process. These parameters were used to generate various results and correlations plots to help better understand the overall heat transfer mechanism. The results show that the convective heat transfer coefficient increased very quickly at the early stage of melting and the rate of increase gradually decreased and became almost steady in the latter stage of melting. The correlation between the Nusselt ($Nu$) and the Rayleigh ($Ra$) numbers show similar trends with other reports. A new correlation for the Nusselt number and Rayleigh number is proposed to characterize the heat transfer during the melting phase with a U-shaped heat source. These data could be attributed to better understand the heat transfer phenomena during melting of a solid PCM and expected to help improve in designing a PCM based thermal energy storage system.

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References:


Chapter 4

4 Investigation on the heat transfer enhancement in a phase change material using nanoparticles

4.1 Abstract

Phase change materials (PCMs) used as a latent heat energy storage media has a drawback of having a low heat transfer rates during melting and freezing processes due to their inherent low thermal conductivity. A commercially available paraffin wax was used as the PCM in this study. Four different nanoparticles were used to see the effect of heat transfer in various nanoparticle-PCM mixtures. The magnitudes of heat flux for PCMs containing each type of nanoparticle were observed to increase between 7-20% compared to PCM only. Also the average time required to reach complete melting was reduced by 30% than PCM only. Among these four nanoparticles used, copper oxide (CuO) was selected as the thermal conductive enhancer for further study. CuO nanoparticles were also added in various mass fractions of 1, 3, 6, 8 and 10% with the PCM. The results indicate that the convective heat transfer coefficient in the melted PCM domain increased gradually to a certain mass fraction of this CuO nanoparticles and then declined. PCM containing 6% CuO nanoparticles resulted in the highest heat flux of 9880 W/m² and showed the best thermal performance. The result also showed that the heat flux in the PCM increased by almost 12% when the CuO mass fraction was increased from 1% to 6%. Also the PCM-6% CuO mixture resulted 25% more melting compared to PCM only. Heat flux was increased between 13-26% when the HTF inlet temperature was increased from 125 °C to 200 °C. The enhancement in the thermal performance in PCM, as a result of adding nanoparticles, reveals their high potential for diverse thermal energy storage applications.

Keywords

Thermal conductive enhancers, nanoparticle, PCM, thermal energy storage, heat transfer fluid, heat transfer coefficient
4.2 Introduction

Thermal energy storage plays a crucial role in ‘holding’ or ‘storing’ thermal energy to match between the demand and supply and, in events when it is necessary to store excess energy that would have otherwise been wasted. A well-designed thermal energy storage system not only improves the performance and reliability of energy systems but also plays an important role in conserving energy. This is an important issue concerning the renewable thermal energy systems and the cogeneration systems. Thermal energy storage methods are classified mainly in three groups: sensible, latent and thermochemical heat storage [1]. Latent heat thermal energy storage (LHTES) systems that use phase change materials (PCMs) as the storage medium have proven to be a better engineering option and are becoming increasingly attractive because of their various advantages such as high energy storage density (energy storage per unit mass), ability to operate within a narrow temperature range around the phase transition temperature of the PCM, compactness, etc. [2-3]. The high energy storage density is the result of utilizing PCM’s latent heat of fusion/solidification. Since there is a long list of PCMs available (single material and/or in mixture form) with a wide range of thermo-physical properties, it has literally opened countless opportunities and flexibilities to work with PCMs. Numerous PCMs with their properties, advantages and limitations have been comprehensively reported in the literature [2, 4]. For all these reasons, PCM have been a very familiar topic in research field for the last three decades [4-6]. It has been reported that there are over 20,000 substances with the melting point in the range 10-90 °C suitable for low to medium temperature applications [7].

Although PCMs are excellent candidate in an energy storage system, most PCMs suffer from a common problem of naturally having low thermal conductivity, which hinders the heat transfer process in a PCM based thermal energy storage (TES) by prolonging the charging (heat addition) and discharging (heat rejection) periods [8]. This also lowers the performance of a TES to a great extent. The thermal conductivities of organic and inorganic materials usually swing around approximately 0.2 W/m/K and 0.5 W/m/K, respectively [9]. Thus, to integrate PCMs in the LHTES, a careful selection of storage media is extremely important.
Due to the low thermal conductivities of the PCMs, some methods or measures are essential that can result in thermal conductive enhancement in these materials for better performance. Significant efforts have been made to overcome this challenging issue. Some of the key approaches utilized are, (i) Extended surface/fins inside the PCM domain, (ii) Imbedding conductive matrix/foam inside PCM, (iii) Encapsulation of PCM, and (iv) Mixing PCM with nanoparticles.

Use of extended heat transfer surfaces (i.e. fin) attached to the heat source is a well-known means for augmenting heat transfer [10]. Meticulously designed fins render superior benefit by extending the effective heat transfer surface area for conduction. In addition, the fin design often induces turbulence and flow mixing which enhances the convective heat transfer coefficient. However, the extended surface also has several demerits especially the increased weight and volume of the storage system. The complexity from the maintenance point and supplemental costs are also of added concerns.

Another option to overcome the barrier of low heat transfer is to employ metal/non-metal foams within the PCM domain, e.g. copper or carbon matrix [11-16]. Such foam/matrixes are composed of very thin strands of material fused together and have been used often in compact heat exchangers [17, 18]. With high thermal conductivity and the presence of ligaments, metal foams act like multiple conducting paths that quickly carry the heat from a higher temperature region to a lower thermal potential region. It has been shown that the solid/liquid phase change heat transfer performance can be improved by 3 to 10 times depending on the metal foam microstructure [19]. On the other hand, non-metals, for example graphite foams have also shown significant improvement in thermal conductivity similar or even better than metal foams [20]. Similar to the extended surface, use of foam in heat transfer augmentation has the major problems of extra weight and volume, cost and obscure maintenance process which limit their use.

Encapsulation of PCMs is another option for enhancing heat transfer. Encapsulations can be of two types: macro or micro encapsulations, depending on the size of the encapsulations. PCM encapsulation with different geometries has its own merits and demerits. Besides flexibility in handling, encapsulations can provide strength, flexibility,
thermal stability and can prevent corrosion and leaks. It acts as a protective barrier for the PCM from harmful interaction with the environment and can provide sufficient surface area for heat transfer. But on the other hand, unless the encapsulation has high thermal conductivity, the system suffers from low heat transfer rate issues. The cost of microencapsulation is also higher compared to other storage methods [21].

The recent development in nanotechnology has opened a new era to enhance the thermal conductance of generally low conductive PCMs in the TES system. Use of heat conductive nano-particles is getting huge momentum and hence particularly, the thermal conductivity of nanoparticles has become a hot field in recent years [22-25]. Due to very small size, nanoparticles exhibit novel material properties, which largely differ from the bulk properties [26, 27]. Nano-sized metallic particles show mechanical, electrical and chemical properties which differ substantially from the bulk material (e.g. higher specific area, lower melting temperature, greater mechanical strength and, different optical, magnetic and electronic properties) that proved attractive in various applications. The heat transfer in presence of nanoparticles is favored since, as the size decreases, the ratio of the surface to volume increased [28-30]. Hence, the thermal conductivity of a fluids can drastically enhance when mixed with nanoparticles even at very low volume fractions [31-34]. Such fluids are called nanofluids [35].

In a review paper, Wenhua et al. [36] mentioned eight parameters observed from different experimental results which effect the thermal conductivity enhancement of nanofluids. These parameters are: particle mass/volume concentration, particle material, particle size, particle shape, base fluid material, temperature, additive, and acidity. It has been reported that mere few percent of nanoparticles in the fluid can significantly increase the effective thermal conductivity of nanofluids [37]. These effective enhancements in various nanofluids have been reported to vary 10-40% [38, 39]. The physical mechanism responsible for the thermal conductivity enhancement of nanofluids is still not well understood [36]. It has also been reported that in general, the thermal conductivity of nanofluids is more temperature sensitive than that of the base fluid. Consequently, the thermal conductivity enhancement of nanofluids is also rather temperature-sensitive. From the studies it was observed that the strong temperature dependence of nanofluid thermal
conductivity is due to the motion of nanoparticles [36], which signifies the importance of convective flow in the heat transfer mechanism. Nanoparticles acts as enthalpic carriers and hence, the heat transfer enhances significantly.

Choi et al. [35] developed a correlation between the heat transfer coefficient and the thermal conductivity of the base fluid \((bf)\) and nano fluid \((nf)\). They also used the same assumptions as that of Dittus and Boelter [40] to derive a Nusselt number correlation for the nanofluids.

Pak and Cho [41] used \(\text{Al}_2\text{O}_3\) and \(\text{TiO}_2\) nanoparticles of mean diameter 13 and 27 nm respectively and a 3 vol\% concentration in a water based nanofluids to determine heat transfer and frictional effects of these nanoparticles under turbulent flow conditions. They observed a substantial increase in the viscosity of nanofluids with increase in nanoparticle concentration. They proposed another Nusselt number correlation for dilute dispersions of nanoparticles similar to the Dittus–Boelter correlation [40]. Using the transient hot-wire method, Lee et al. [42] experimentally measured thermal conductivity of nanofluids prepared from CuO nanoparticles (23.6 nm mean diameter) in ethylene glycol (EG) and showed that the thermal conductivity for the CuO-EG nanofluid can be enhanced by more than 20% with a nanoparticle volumetric concentration increased up to 4%.

Wang et al. [43] proposed a model to determine the effective thermal conductivity of nanofluids. They compared results from their model to experimental data in a nanofluid comprising of CuO particles (50 nm) in DI (deionized) water with a volumetric concentration of less than 0.5%. They argued that beyond this dilute limit, the model needs to be refined by taking into account the effects of the possible deposition of the particles. Fotukian and Esfahany [44, 45] experimentally investigated the turbulent convective heat transfer in a CuO nanofluid with particle volume concentration 0.3%, where they observed the heat transfer coefficient to increase by 25% compared to pure water.

Some studies reported the effect of nanoparticles’ concentration on the viscosity of nanofluid. Namburu et al. [46] conducted measurements of viscosity of CuO nanoparticles dispersed in 60:40 EG/Water solution. They found variations of the viscosity with changes in concentration and temperature and proposed various correlations. To maintain a
moderate viscosity increases, oxide based nanoparticle concentrations are normally kept below 5% by volume which causes a thermal conductivity enhancement level typically up to about 30%. At higher particle volume concentrations, nanofluid viscosity would increase, consequently the increase in the thermal conductivity enhancement is reduced or even reversed [36]. Hence, it is understood that for each base fluid-nanoparticle combination, depending on the operating conditions, there is a limit on how much benefit is achievable through the increase in the nanoparticle concentrations.

The above review shows that the use of nanoparticles to enhance heat transfer rate in a PCM based thermal storage system is promising and it can be considered as the best option among all conductive enhancing techniques. However, the literature review indicated, that the nanoparticles were primarily used in single phase fluids, and very limited studies reported the use of nanoparticles in the PCMs for thermal storage applications. There is scarcity of studies investigating the effect of nanoparticles in a PCM on the underlying heat transfer during the phase change process.

The present study is focused on a detailed investigation of the influence of nanoparticles in a PCM on the transient heat transfer during the melting process. Different types of nanoparticles were considered and the detailed heat transfer analysis was performed.
4.3 Experimental setup

4.3.1 Experimental setup and the PCM Chamber

The schematic of the experimental setup used in this study is shown in figure 4.1. A rectangular shaped chamber is used as the PCM storage (figure 4.2). The chamber has the inner chamber dimensions of 100 mm long (L), 100 mm high (H) and 12 mm deep (D). It is made from an 8 mm thick optically clear cast acrylic plate cut into a U-shaped configuration with a width of 25 mm in all sides. The front and back walls of the chamber were formed using a 5 mm thick plain clear glass sheet and an acrylic sheet of same thickness, respectively. Both sheets were 150 mm long and 125 mm high. Two, 2 mm thick rubber gaskets were placed between the U-shaped acrylic plate and the front and back walls for leak prevention and to minimize heat loss.

4.3.2 Heat source

A copper tube with an outer diameter (OD) of 4.76 mm and a wall thickness of 0.355 mm was inserted horizontally into the PCM chamber near the bottom of the storage chamber through the 8 mm acrylic plate via two holes. Hot air was passed through this copper tube that served as the heat source. The total length of the tube passing through the chamber and in contact with the PCM was 100 mm. To prevent direct contact of the tube with the acrylic side plate, two circular rubber seals were placed over the tube circumference in places where the tube was in contact with the chamber walls. Purpose of the rubber seals was to reduce the heat transfer between the heat source and the acrylic plate as well to prevent any leakage of the PCM from the storage chamber when it was melted. Rubber seals were used between the tube and the chamber walls to prevent a direct contact of the tube with the acrylic block. These rubber seals not only minimized the heat transfer between the tube and the acrylic plate, but also restricted any leakage of the liquid PCM from the storage chamber. All parts of the setup except the front glass pane were insulated to avoid heat losses. The storage chamber was attached vertically onto a metal frame.
Figure 4.1: Schematic diagram of the experimental setup. The horizontal heat source (HTF tube) is shown near the bottom of the chamber

4.3.3 Controlling temperature of the heat source

The hot air (hereinafter termed as the heat transfer fluid, HTF) was heated while passing through an air heater (Zesta ZA0017) equipped with a PID controller (Zesta ZEL-9100) to control its temperature. The accuracy of the PID controller was within ±2°C. As seen in the experimental setup in figure 4.1, the compressed air from the supply line was first passed through a settling tank to dampen any pressure fluctuations. A pressure regulator was connected to control the pressure of air exiting the settling tank. The air was then passed through a flow meter to control its flow rate before passing through the heater. The flow rate of air was kept between 30-40 liters/min. The accuracy of the flow meter was estimated as ±5%. Four different input heat loads at the chamber inlet were applied by varying the temperature of the HTF through the heater controller to 125 °C, 150 °C, 175 °C and 200 °C.
Figure 4.2: Schematic of the PCM chamber, (a) Isometric view, (b) locations of the thermocouples are marked with open circles
4.3.4 Thermocouples for temperature data acquisition

In the present study, a total of 31 thermocouples (TCs) were employed to measure the temperatures at various locations. These thermocouples were T-type (T30-2-507, Zesta Engineering) with an accuracy of ±0.4%. The locations where thermocouples were placed in the apparatus are shown with black open circles in figure 2(b). Two of these thermocouples were employed to measure the temperatures of air (HTF) at the inlet and outlet of the PCM chamber, which was used to compute the input heat flux. The HTF tube surface temperature inside the chamber was measured at five different locations along the tube length with equal spacing of about 20 mm between them. The first and last thermocouples on the tube surface were located 8 mm from the chamber inlet and outlet, respectively. For temperature measurements within the PCM domain, 20 thermocouples were arranged in an array format of 4 rows and 5 columns (figure 4.2). The distance between these thermocouples was also kept at about 20 mm while, the first row was located 15 mm above the HTF tube. To measure the temperature of the PCM very close to the HTF tube surface, four thermocouples were placed at 1, 2.5, 5 and 10 mm above the HTF tube surface at the mid axial location (figure 2). All thermocouples were inserted into the chamber through the insulated acrylic back wall (figure 2a).

The temperature data from all thermocouples were recorded simultaneously via data acquisition cards (thermocouple modules: NI-9214 and NI-9211) controlled by the LabVIEW software, at a rate of 3 Hz. The temperature data recording was started 1 min prior to the beginning of the heating process and continued 5 min after the PCM was completely melted.

4.3.5 Thermocouple calibration

Calibration of the thermocouples was conducted by placing them in an insulated hot water bath and using a high accuracy thermometer (Kessler 10/30 ST Jointed Thermometer) which served as the reference temperature. The calibration was conducted over a temperature range from 90 °C to 0 °C with approx. 10 °C temperature increments.
Calibration curves showed almost linear relationship. The uncertainty of thermocouples was ±0.4%.

**4.3.6 Phase Change Material (PCM)**

A commercially available paraffin wax known as ‘Polyfin’ with an average melting point of 55°C and a specific gravity of 0.92 in solid phase was used as the phase change material (PCM) in this study. The volume change of the wax in the liquid state was estimated to be approximately 10% from the solid state. This PCM was chosen primarily due to the reason that the wax changes from its white color in the solid phase to transparent form in the liquid phase after melting. This enabled to clearly distinguish the two phases during melting via imaging and track the solid-liquid interface.

**4.3.7 Thermal Conductive Enhancers (TCE)**

As mentioned earlier, a variety of nanoparticles have been used with base fluids and a significant variation in their performances has also been reported in the literature. Hence, selection of the appropriate nanoparticle for a given base fluid or a given application is not an easy task. Some key factors were considered to select nanoparticles for the present study. Firstly, the frequently used nanoparticles were identified from various studies similar to the present study. This gave a better understanding and a scope for possible comparison with present study. Secondly, few trial runs were performed with few types of nanoparticles at various mass fractions to identify potential nanoparticles for the present study. Thirdly, the viability of such choice in practical applications was also considered since, other than the thermo-physical considerations, cost of nanoparticle is one crucial factors that limit its use in large scale real applications. By considering these factors, the choice for nanoparticles was narrowed down and initially four nanoparticles were selected for the present study, which were 1) aluminum oxide, Al₂O₃ (20 nm), copper oxide, CuO (40 nm), silver (30 nm) nanoparticles and multi-walled carbon nanotubes. MWCNT (diameter 40-60 nm, length 2 μm). Various properties of these nanoparticles are presented in Table 4.1.
<table>
<thead>
<tr>
<th>Nanoparticle</th>
<th>Size, Nanometer (nm, 10^-9 m)</th>
<th>Purity</th>
<th>Bulk density, gm/cc</th>
<th>True density, gm/cc</th>
</tr>
</thead>
<tbody>
<tr>
<td>γ-Al2O3 (Alumina)</td>
<td>20</td>
<td>99+%</td>
<td>&lt;0.1</td>
<td>3.89</td>
</tr>
<tr>
<td>Silver (Ag)</td>
<td>30</td>
<td>99%</td>
<td>0.312</td>
<td>10.50</td>
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<td>CuO (Copper oxide)</td>
<td>40</td>
<td>99.99%</td>
<td>0.790</td>
<td>6.40</td>
</tr>
<tr>
<td>MWCNT (Multi-walled carbon nanotube)</td>
<td>Diameter 40-60, length, 2 µm</td>
<td>99+%</td>
<td>0.220</td>
<td>~2.18</td>
</tr>
</tbody>
</table>

### 4.3.8 Preparation of PCM-nanoparticle mixture

Nanoparticles were mixed in various mass fractions with the paraffin wax (PCM) when the latter was in solid state (as pellets). The mixture was then melted on a hot plate and the liquid composite was continuously stirred for 72 hours by maintaining the temperature around 65-75 °C. The problem with most of the thermal conductive nanoparticles, whether it is a metal (e.g. Ag nanoparticle), metal-oxide (e.g. CuO) or a non-metal (e.g. MWCNT) is that, they are much heavier than the typical base fluid [47]. Keeping nanoparticles suspended is a big challenge. In addition to this problem, the nanoparticles always try to agglomerate to form a cluster, resulting in a heavier concentration of solids in places which influences the heat transfer [49]. Use of some methods like sonication in a hot bath or apply some suitable dispersant or surfactant can help improve the issue [36, 50]. Unfortunately, these issues have not been well addressed in the literature. In the present study, Tween 60, a commercially available neutral dispersant/surfactant was added to the PCM-nanoparticle mixture (approximately 10% wt. of nanoparticle) to improve the submersion ability of the nanoparticles in the nanofluid.

### 4.3.9 Heat loss from the PCM chamber
To facilitate the imaging, a clear glass pane was used as the front wall which was not insulated. The upper end of the chamber was left open to room atmosphere to accommodate thermal expansion of the PCM during melting. To determine the heat loss from the front glass pane and whether this heat loss had any effect on the underlying melting process and associated heat transfer within the PCM domain, a separate set of experiments was conducted at the same setting of all input parameters after insulating the front wall with the same insulation. Comparisons were conducted between the two sets of experimental results corresponding to insulated and uninsulated front glass pane. Pure wax with no nanoparticles was considered for this comparative study. The results show a slightly higher temperature inside the PCM domain for fully insulated case than that of the case with the front glass pane, as expected. Comparison of results at different locations within the domain shows that the difference in the mean temperatures for the two cases was on average of 8.7%. Since the process was transient by nature, the temperature profiles at different locations within the domain over the entire melting period were compared for both cases to see the effects of the insulation. The results show almost similar trends for both insulated and uninsulated cases which confirmed that the uninsulated pane did not have any considerable influence on the underlying transient melting behavior of the PCM.

4.3.10 Image acquisition

The melting process of the PCM was imaged through the front panel by using a high resolution 4 MP CCD camera (Flare 4M180-CL) with a resolution of 2048 × 2048 pixels. An image acquisition system (DVR express core, IO Industries) was used to record images via the DVR express CoreView software installed on a PC. The images were then transferred to an external storage media for further processing. The camera field of view of 94 mm × 94 mm was set for all experiments. The image acquisition started at the beginning of the melting process and continued until the PCM was completely melted. For all experiments, the imaging rate was set at 0.1 Hz i.e. one image every 10 seconds.

The images acquired from the camera were processed using an in-house Matlab code to detect solid-liquid interface as well as the melted fraction in a transient manner. The code executed image processing routines to detect the interface based on the variations in the
gray-scale values through the solid and liquid phases. By performing a series of morphological operations the noise from the data was removed to improve the accuracy of the interface detection. The coordinates of the detected interface were then recorded at every spatial location. An example of the image acquired during experiments along with the corresponding detected interface is shown figure 4.3. The white region in the image represents the solid phase and the dark gray region represents the liquid wax. The detected interface is plotted with the red color. The uncertainty in detecting the interface was within ±2 pixels that correspond to ±90 µm. By processing a series of consecutive images from the beginning to the end of the melting process using this procedure, spatio-temporal movement of the solid-liquid was obtained for each case. This information was used in the further analysis of the data. Note that the interface movement in the immediate vicinity of a thermocouple was locally disturbed. Hence, the interface data in the vicinity of each thermocouple were excluded from subsequent analyses.

![Image](image.png)

Figure 4.3: Instantaneous image showing the PCM chamber domain during melting. The dark grey and white regions represent liquid and solid domains of the PCM, respectively

4.4 Computation of various thermal parameters
The total heat loss by the HTF passing through the chamber was calculated based on the hot air inlet and outlet temperatures and the flow rate, using the following equation:

\[ \dot{q} = \dot{m}c_p(T_{in} - T_{exit}) \]  

(1)

where, \( \dot{q} \) is the heat transfer rate, \( \dot{m} \) is the mass flow rate of air, \( c_p \) is the specific heat of air, and \( T_{in} \) and \( T_{exit} \) are the inlet and exit temperatures. Since the thermocouples to measure inlet and outlet temperatures were located 1.5 cm off from the physical entrance and exit of the PCM storage, the heat loss from the HTF tube between \( T_{in} \) and \( T_{exit} \) thermocouple locations and the chamber inlet and outlet locations, respectively, were calculated. The net heat transferred to the PCM (\( \dot{q}_{net} \)) was then calculated by subtracting the inlet/outlet heat loss, from the total heat lost by the HTF. The net heat flux (\( q''_{net} \)) into the PCM domain was computed by dividing (\( \dot{q}_{net} \)) by the surface area of the HTF tube inside the domain. The uncertainty in estimating the heat flux was approximately on average within ±10%. (see appendix C).
4.5 Results and Discussion

Data acquired simultaneously from the temperature and the image recording systems, provided a detailed insight into the thermal behavior and the dynamics of solid and liquid transition during the phase change process. The results are primarily focused on the investigation of the influence of nanoparticles and their volume fractions in the PCM on the melting behavior and associated heat transfer.

4.5.1 Selection of thermal conductive enhancer (TCE)

As mentioned earlier in the experimental setup section, four different thermal conductive enhancers (TCE) in the form of nanoparticles were considered in this study to investigate their influence on the melting behavior and the associated heat transfer, which were, Aluminum Oxide or Alumina (Al$_2$O$_3$), Copper Oxide (CuO), Multi walled carbon nanotube (MWCNT), and Silver (Ag).

These nanoparticles were mixed with the PCM to produce separate samples. For comparison purpose, the inlet conditions of the HTF fluid (hot air) were maintained at the same settings for testing all nanoparticles’ samples as well as the reference case i.e. plain wax (PCM with no nanoparticles). For all experiments, the HTF flow rate was kept at 40 litres/minute at the inlet pressure of 15 pisg and the HTF inlet temperature was maintained at 150 °C.

4.5.2 Nanoparticles floatation/suspension

Since most of the high conductive nanoparticles are heavier than the PCM used in the study, the holding of the particles suspended in the liquid was a difficult task and a big challenge. The sample tests showed that the sinking rates of Al$_2$O$_3$ nanoparticles and MWCNT were more than that of CuO and silver nanoparticles. The sinking/flotation problem with MWCNT is probably due to its inherent nature for particle aggregation, which might have resulted in the formations of clusters of MWCNT [49, 51]. Thus, bigger
chunks of MWCNT nanoparticles were formed and sank quickly due to weight. This was evident by the amount of particles deposited on the surface of the HTF tube as viewed from the front glass during melting. This issue was the most significant for Al₂O₃ particles, whose deposition on the HTF tube is clearly visible in figure 4.4. This deposition of particles on the tube is expected to create additional thermal resistance. Alumina (Al₂O₃) being a metal oxide is naturally less thermal conductive than the bare copper tube and hence, the particle deposition on the tube surface negatively affected the heat transfer rate. Al₂O₃ is by nature highly hydrophilic [48] in contrast to paraffin wax which is highly hydrophobic or rather more oleophilic [50]. Since, any substance which is hydrophilic (e.g. Al₂O₃), by nature does not interact well with substances which are oleophilic (wax in the present case). This explains why Al₂O₃ was sinking at a higher rate than the CuO and silver nanoparticle.

Figure 4.4: Image showing deposit of Al₂O₃ nanoparticles (NP) as white powder like substance on the surface of heat source in two different stage of the melting. Rate of nanoparticle deposit is seen to have increased as the melted volume increased
4.5.3 Temperature profiles

As mentioned earlier, temperatures were measured at 24 locations inside the chamber to characterize the thermal behavior of the PCM during the melting phase (see figure 2). In the mid-axial plane of the PCM domain, thermocouples were located at eight vertical locations (1, 2.5, 5, 10, 15, 35, 55 and 75 mm from the top surface of the HTF tube. The time history of the PCM temperature at four of these vertical locations (2, 10, 35, 55 and 75 mm) are plotted in figure 4.5 for different nanoparticles’ mixtures with the wax (1% mass fraction for each nanoparticle) and the reference plain wax (from Chapter 2), for the same HTF input conditions. From all four graphs, it is clearly evident that at a given time during the melting process, the temperatures of the PCM in the domain in the presence of nanoparticles are significantly higher than that with the plain PCM. Hence, at a given location, the PCM reaches its melting state much faster in the presence of nanoparticles compared to the plain PCM, which implies a faster heat transfer rate in the former case. It is observed that on average, the PCM in the presence of nanoparticles reached melting state in 40% less time than that without nanoparticles up to mid-height of the chamber. At greater heights, the time difference is reduced to about 25%. Comparison of the results among PCM-nanoparticle mixture show that PCM-silver nanoparticle mixture exhibited relatively the highest temperature throughout the melting process and continued after melting. While alumina showed the lowest temperature among nanoparticles. The performances of CuO and CNT were generally in between silver and alumina particles.
Figure 4.5: Comparison of the time histories of the PCM temperature at different vertical locations: a) 2 mm, b) 10 mm, c) 35 mm, d) 55 mm and (e) 75 mm with four different types of nanoparticle at 1% mass fraction, HTF inlet temperature: 150 °C, HTF flow: 40 litres/min
4.5.4 Effects of nanoparticles on heat flux ($q''$)

The results in figure 4.5 showed a faster rate of heat transfer into the PCM domain due to the presence of nanoparticles. In order to quantify this behavior, the net heat flux into the PCM domain ($q''_{net}$) was computed for each case. The results are presented in Table 4.2. The results show as expected, that the net heat flux into the storage chamber is higher in the presence of nanoparticles. The magnitude of the heat flux increase relative to that for the plain wax varied approximately between 7-20%. The results show the highest heat flux increase for silver nanoparticles and lowest for alumina nanoparticles. These results further confirm and support previous observations that nanoparticles increase the heat transfer rate.

<table>
<thead>
<tr>
<th>PCM</th>
<th>$q''_{net}$, W/m²</th>
<th>% increase in $q''$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain Wax</td>
<td>8150</td>
<td>---</td>
</tr>
<tr>
<td>Silver, 1%</td>
<td>9720</td>
<td>19.3</td>
</tr>
<tr>
<td>CuO, 1%</td>
<td>8750</td>
<td>7.4</td>
</tr>
<tr>
<td>MWCNT, 1%</td>
<td>9200</td>
<td>12.9</td>
</tr>
<tr>
<td>Al₂O₃, 1%</td>
<td>8710</td>
<td>6.9</td>
</tr>
</tbody>
</table>

4.5.5 Effects of nanoparticles on heat transfer coefficient ($h$)

During the phase change process both conductive and convective heat transfer modes exists in the domain in solid and liquid phases, respectively. As the fraction of one phase continues to increase while the fraction of the other phase continues to decrease, the magnitudes of corresponding heat transfer modes also change continuously throughout the phase change process. Hence, the heat transfer process in a PCM during phase change is more complex than that in a single phase material. Since the convective heat transfer is
dependent on the flow behavior, due to the continuous change in the liquid fraction throughout melting, the liquid flow behavior is expected to change, which in turn influences the heat transfer. Hence, the convective heat transfer coefficient \((h)\) is expected to change in a transient manner during the phase change (melting) process.

The convective heat transfer coefficient \((h)\) was computed using the Newton’s law of cooling equation [52] as:

\[
h = \frac{q''_{\text{conv}}}{(T_s - T_b)}
\]

where, \(q''_{\text{conv}}\) is the convective heat flux, which based on the energy balance is set equal to \(q''_{\text{net}}\), and \(T_s\) and \(T_b\) are tube surface and bulk liquid PCM temperatures, respectively. As the liquid fraction continued to increase with the progression of melting, the local heat transfer coefficients were computed when the melted fraction reached a given thermocouple. For the case of a thermocouple row, the local heat transfer coefficient was computed only when the melted region reached all five thermocouples in that row. Hence, eight values of the local heat transfer coefficient were computed as a function of the melted fraction (figure 4.6). Note that the bulk liquid temperature was computed as the average from all those thermocouples that were in the liquid domain.

The local (transient) heat transfer coefficient is plotted versus the liquid fraction in figure 4.6 for different PCM-nanoparticle mixtures. The values of local heat transfer coefficient for the plain PCM case are also plotted for comparison. The results show a similar trend for all cases i.e. the value of \(h\) increased more rapidly in the early stages of the melting process and then became more gradual. Comparison showed that overall, CuO-PCM mixture has the highest heat transfer coefficient followed by the silver-PCM mixture. The values of \(h\) for both of these cases are significantly higher than the reference (plain PCM) case and other two nanoparticles. On average the values of \(h\) for CuO and silver nanoparticles cases were 13.5% and 17.5% higher than the case with no nanoparticles. The higher value \(h\) for these two cases indicate that these thermal conductive enhancers also influence the flow behavior in the melted (liquid) PCM region and likely contributed to the enhancement of fluid mixing. The visual observations of the flow behavior through
imaging show that the rate of sinking of these two nanoparticles with the progression of melting was lower than the MWCNT and Al₂O₃ nanoparticles. The values of $h$ for MWCNT and alumina cases were almost the same as the reference case. The results show no improvement of $h$ for MWCNT and alumina cases. A plausible explanation for this behavior is the higher sinking rates of these particles which most likely tend to counter the enhanced mixing trends. CuO and silver nanoparticles, although heavier than alumina, were observed to have much lower rates of sinking likely due to better affiliation with liquid PCM than Al₂O₃ as mentioned earlier [48-50].

Figure 4.6: Convective heat transfer coefficient ($h$) as a function of melted fraction for plain wax (PCM) and various PCM-nanoparticle mixtures. All nanoparticles were mixed at 1% by wt. of the PCM. The temperature and flow rate of HTF at inlet were 150 °C and 40 litres/min respectively.
4.5.6 Temperature profiles with different mass fractions of CuO nanoparticles

In the preceding section, various nanoparticles (with the same mass fraction and under the same HTF input conditions) were considered to evaluate their performance as thermal conductive enhancers in a PCM during the melting process. In order to further investigate the impact of other parameters such as mass fraction and the heat flux, the best among these nanoparticles need to be selected. The results show that CuO and silver nanoparticles exhibited better performance than the other nanoparticles. Considering economic factor, which is very critical from practical aspects, CuO nanoparticles are much more cost effective than silver nanoparticles. Hence, CuO nanoparticles were selected for further parametric study.

To investigate the effect of mass fraction on the thermal performance of the mixture, five samples of PCM-CuO nanoparticle mixtures were prepared with 1, 3, 6, 8 and 10% mass fractions of CuO in the wax. The experiments were conducted for each mixture at the inlet HTF conditions of 150 °C and 40 liters/min. The time history of temperature variation in the PCM mixtures with different nanoparticle mass fractions are shown in Figure 4.7 at three heights in the chamber. The influence of the mass fraction of CuO is more clearly evident closer to heat source (Figure 4.7a), where it is observed that the melting rate increased with an increase in the mass fraction of CuO up to about 6% and then it started to decrease. This trend started to become less prominent with an increase in height within the PCM domain. At a height of 10 mm from the heat source, the melting time of the PCM mixture with 6% CuO mass fraction was about 30% shorter than that for 1% CuO mass fraction. At a height of 35 mm and 55 mm from the heat source, the melting time percentages relative to 6% CuO mass fraction changes to 12% and 7% for 1% CuO mass fraction.
Figure 4.7: Temperature profiles of PCM with five separate mass fractions (1, 3, 6, 8 and 10% respectively of CuO nanoparticles at two vertical locations, a) 10 mm, b) 35 mm and c) 55 mm with the HTF input temperature and flow rate of 150 °C and 40 litres/min. The profile for the plain PCM was also plotted for comparison.
The influence of the mass fraction of CuO nanoparticles on the melting rate was further investigated by comparing the melted fraction versus time for PCM mixtures with different CuO mass fractions. The melted fraction of the PCM at different instances of time was computed by analyzing the images of the PCM domain during the phase transition using an in-house image processing algorithm. The results for the four different CuO nanoparticle-PCM mixtures are shown in Figure 4.8. The results for the plain PCM are also plotted for comparison. As expected, the melting rate of the PCM mixed with nanoparticles was higher than that for the plain PCM. The results also confirmed that throughout the process, the melting rate was highest in the PCM with 6% CuO mass fraction. The results also indicate that on an average approximately PCM with 6% CuO was melted 25% more compared to the plain PCM during the same time period.

![Figure 4.8: Rate of melting with various mass fractions of CuO nanoparticles in the PCM. Melting rate of plain PCM is also plotted for comparison.](image)

The effect of the mass fraction of CuO nanoparticles was also evaluated by computing the heat flux into the PCM domain for each condition. The results are presented in figure 4.9. As observed from the figure, the computed heat flux into the PCM increased with an
increase in the CuO mass fraction up to about 6% and then started to decrease, which indicate that 6% CuO mass fraction in the PCM is an optimum fraction. The results also show that the heat flux into the PCM increased by almost 12% when the CuO mass fraction increased from 1% to 6%.

![Figure 4.9: Effect of CuO nanoparticle mass fractions on heat flux (q’’)](image)

4.5.7 Effects of the variations in the input heat load on the PCM temperatures with 6% CuO nanoparticles

As results in Figures 4.7 and 4.9 show, 6% mass fraction of CuO nanoparticles in the PCM provide the best thermal performance. The effect of the inlet HTF conditions on the thermal performance of 6% CuO-PCM mixture was also investigated. The experiments were conducted for three other inlet HTF conditions that corresponds to the HTF inlet
temperatures of 125 °C, 175 °C and 200 °C at the same flow rate of 40 litre/min. The time history of the PCM temperature at different vertical locations (2, 10, 35 and 55 mm) is shown in Figure 4.10 for all four heat fluxes. The plots show that the rate of temperature change is a function of both spatial location and the heat flux. Near the heat source, the temperature rise starts at smaller values of $t/t_s$, and gradually reaches to $t/t_s = 1$. The effect of heat flux is more dominant in the early state of heat transfer. At lower heat fluxes, the temperature rise is gradual from the beginning of the heating process until $t/t_s = 1$. As the heat flux increases, the temperature remains relatively uniform in the early stages of heating and then sharply rises to the quasi-steady state. The plots also show that an increase in the distance from the heat source, extends the time range of the initial near-uniform temperatures at high heat fluxes. For HTF input temperatures of 175 °C and 200 °C, as examples, the near-uniform temperature range extends to approximately $t/t_s = 0.7$. It is also observed that the rate of temperature rise becomes almost independent of the heat flux at higher heat flux magnitudes away from the heat source.
Figure 4.10: 6% CuO-PCM temperature in non-dimensional form versus the normalized time at four heat fluxes and four different heights from the heat source. (a) 2.5 mm, (b) 10 mm, (c) 35 mm and (d) 55 mm. $T_m=$PCM melting temperature, $t_s=$ time when PCM temperature became steady at the given location.

4.5.8 Comparison for heat flux between plain wax and PCM-nanoparticle mixture for 6%wt. CuO

Figure 4.11 shows the heat flux into the 6% CuO-PCM mixture for different inlet HTF temperatures. The results at the same inlet HTF temperatures for the plain wax are also plotted for comparison. The heat flux into the PCM increased with an increase in the inlet HTF temperature, as expected. The results also show that the heat flux into the PCM was higher in the presence of nanoparticles compared to the plain PCM. However, the percentage increase in the heat flux in the presence of nanoparticles was strongly dependent on the input HTF temperature. It is observed that at the lowest inlet HTF temperature of 125 °C, the computed heat flux into the PCM (6870 W/m²) was approximately 13% higher in 6% CuO-PCM mixture when compared with plain PCM, which increased with an increase in the HTF inlet temperature. At the highest inlet HTF temperature of 200 °C, the computed heat flux (14,500 W/m²) into the PCM was approximately 26.5% higher for 6% CuO-PCM mixture compared to plain PCM. A plausible cause for this dependency on the
HTF temperature is that, as the temperature of the PCM was increased, the viscosity of the nanoparticle enriched PCM was decreased which enabled enhanced heat transfer rate [36, 46], since an enhanced and better convective flow in a less viscous nanofluid is natural and expected. It has also been reported that in general, the thermal conductivity of nanofluids is more temperature sensitive than that of the base fluid [36]. Consequently, the thermal conductivity enhancement of nanofluids is also rather temperature-sensitive. Increase in temperature decreased the viscosity and hence, nanoparticle in the fluid was more prominent [36]. These results also indicate that change in effective viscosity of nanoparticle-PCM mixture could be an important parameter to consider since it can limit the effectiveness of nanoparticles in a given PCM based on inlet temperature conditions.

![Graph](image_url)

**Figure 4.11: Effects of the HTF input temperatures on the computed heat flux and comparison with plain PCM**

The influence of the change in effective viscosity of nanoparticle-PCM mixture is also evident in the transient convective heat transfer coefficient values shown in figure 4.12 for
different inlet HTF temperature conditions i.e. different heat flux magnitudes into the PCM for 6% CuO-PCM mixture. The results show that the magnitude of the local heat transfer coefficient increased with an increase in the heat flux. The convective heat transfer coefficient is strongly influenced by the flow behavior within the liquid domain. An increase in the heat transfer coefficient value implies a faster transportation of heat by fluid parcels. Fluid viscosity tends to resist the fluid motion and hence lower mixing and heat transport. As mentioned above the increase in the temperature lowers the effective viscosity of the mixture which increases the fluid movement and enhances mixing, which increases the magnitude of the local heat transfer coefficient.

![Convective heat transfer coefficient (h) as a function of the melted fraction at different heat fluxes for PCM-nanoparticle mixture at 6%wt. of CuO nanoparticle](image.png)

**Figure 4.12:** Convective heat transfer coefficient (h) as a function of the melted fraction at different heat fluxes for PCM-nanoparticle mixture at 6%wt. of CuO nanoparticle
4.6 Discussions and Conclusions

This experimental study was done in a thin rectangular chamber containing paraffin wax, used as the PCM and four types of nanoparticles at 1% mass fractions were added to the PCM to investigate performance of various thermo-physical parameters. All experimental results were compared with a previous study by the same authors (included in chapter 2) with net PCM. From the results of this study, it can be generally concluded that the thermal performance of a PCM based latent heat storage can be improved substantially using thermal conductive enhancers. But the heat transfer enhancement depends on many factors like, particle mass/volume concentration, particle material, particle size, temperature and additives. Additionally, it also depends on thermal conductivity, viscosity, specific heat of the mixture, interaction of the nanoparticles with the base fluid, imposed temperature and the nature of the convective flow and so on.

The results also showed that mass fractions of the nanoparticles have a limitation on effective heat transfer rate. This is due to the fact that adding nanoparticles although improves the conductivity, but simultaneously increases the viscosity of the nanofluid which hampers the convective flow and thus the heat transfer within the fluid. The balance between this two counter effects reaches an optimal value for a certain mass fraction of nanoparticles and beyond which adding nanoparticles would simply reduce the performance of a LHTES system. From the results of this study, it can be concluded that the thermal behaviour of a nanoparticle enriched PCM is helpful for a PCM based latent heat storage which can eliminate or reduce the general problem of low heat transfer rate in a PCM. The careful selection of the nanoparticle and PCM (acting as the base fluid) combination considering the operating conditions and the intended use is extremely important. Because many factors simultaneously play crucial roles, a detail and comprehensive data is required before a final selection can be made.

Acknowledgement

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References:


Chapter 5

5 Conclusions and Recommendations

5.1 Overall Conclusions

Thermal energy storage (TES) is an excellent way to preserve energy in the form of heat. This can be a great tool to alleviate the mismatch between the demand and supply of energy, which is one of the major barriers in the effective utilization of the most prolific renewable energy resource i.e. the solar energy, which is inherently intermittent and unpredictable in nature. Thermal energy storage is also a critical device to recover and reuse waste heat. The thermal energy storage systems utilizing PCM as the storage medium have distinct advantage over sensible energy storage system due to the exploitation of the enthalpy of phase change as the energy storage mechanism. However, the heat transfer process to and from a PCM during phase change is complex due to the simultaneous existence of two phases where, those fractions are continuously changing with time. The complexity of the problem further increases as the orientation and position of the heat source/sink in the storage domain changes. The present dissertation is aimed at improving the fundamental understanding of the dynamical behavior of the solid-liquid phase transition during melting, and a better characterization of the associated heat transfer. To achieve this goal, a comprehensive experimental investigation was undertaken. The present research, comprised of three parts with the focus on convective heat transfer process in the melted domain and the dynamical behavior of the solid-liquid interface.

The experiments were conducted in a thin rectangular chamber served as the storage medium. The chamber was thin enough to assume it as a 2D chamber. Using a set of thermocouples, the temperature measurements at multiple locations were performed to characterize the thermal behavior of the PCM. An imaging technique was also employed to simultaneously acquire images of the moving solid-liquid interface. Commercial paraffin wax was used as the PCM. Hot air passing through a copper tube inside the storage domain was used as the heat source.
In Chapter 2, the investigations were carried out in the PCM storage chamber with a horizontal heat source near the bottom. Results show that the shape of the solid-liquid interface was comparatively linear during the early stage of the melting process but, once the melted fraction increased, it became nonlinear. This trend is attributed to the domination of the local convective cells that immensely influenced the melting rate. Results also show that the magnitude of the solid-liquid interfacial velocity increased with an increase in the input heat flux, as expected. The results also show that the heat flux into the PCM domain significantly affected the melted (liquid) phase, which is also likely due to the strong presence of the convective motions. The simultaneously recorded temperature and the images were analyzed which enabled to characterize various important thermophysical parameters associated with the PCM during the melting process. It was observed from the results that the convective heat transfer coefficient, $h$, increased rapidly until approximately 20% of the PCM volume was melted, followed by a more gradual increment in $h$, which then became nearly steady when the melted fraction reached approximately two-third of the total PCM volume. A new Nusselt number ($Nu$) and Rayleigh number ($Ra$) correlation is proposed that characterizes the heat transfer during the melting phase of a PCM in a rectangular chamber when heated by a bottom heat source and its trend relative to the established correlations in the literature for the closest configurations was also discussed.

In chapter 3, an experimental study similar to chapter 2 was conducted where the melting of a PCM inside a similar chamber was accomplished by using a U-shaped cylindrical heat source which had a different geometry and orientation than the heat source in chapter 2. The purpose of this study is to better understand the effect of the geometry and orientation of the heat source on the melting and heat transfer behaviour in a PCM. The results show that the imposed heat load and the position of the heat source have a significant effect on the profile and orientation of the interface during the melting period. The time history of the temperature profiles provided a better insight into the heat transfer process. Results show that the interface inside the PCM domain progressed in a nonlinear fashion although the interface velocity showed some consistency when moving away from the heat source. The results also show that the heat flux significantly affected the melted (liquid) phase, likely due to the variations in the convective motions. The heat transfer behavior, interface
movement and the heat transfer coefficient differed both axially and vertically as well as with the melting rate. Because of effects of virtually two heat sources, formed by the two legs of the U-shaped cylindrical heat source, the inner region melted quickly than the outer regions. This trend is reflected from the observed results where it shows approximately two-third of the total PCM volume was melted at a quicker rate than the last one-third volume, which is mostly the PCM in the outer region. It is also observed that the local heat transfer coefficients ($h_{local}$) was influenced by the heat source orientation and as such the $h_{local}$ was observed significantly higher in the liquid PCM zones within the inner region than the outer region. The $h_{local}$ in the inner region was observed 35% higher on average than the outer zone when the heat fluxes was increased from 3450 W/m$^2$ to 5840 W/m$^2$. The reason for this trend can be explained as, the volume of the liquid in the inner region was more than the outer region during the same period of melting, length of the HTF tube in contact with the liquid was higher which resulted in a higher magnitude buoyancy-induced flow in the inner region than the outer. Also the increased liquid volume resulted in the growth of strong convective cells, which helped to increase the heat transfer. Similar trends were also observed in the vertical orientation. The buoyancy induced convective currents from the bottom tube surface supported the flow induced by buoyancy from the vertical sections of the tube and played a significant role in the overall convective motion and, resulted in higher Nusselt number values. The results also showed that the Rayleigh number increased almost monotonically with the heat flux in all four zones within the chamber. It is observed that the Rayleigh number on average increased by approximately 23% in the inner zone and 18% for the whole domain as the heat flux increased from 3450 W/m$^2$ to 5840 W/m$^2$. This is because the increased temperature in the inner region is believed to have caused a higher reduction in the viscosity, resulting in stronger fluid motion. A generalized $Nu-Ra$ expression for this relation is proposed using a power law fit to the experimental data.

Chapter 4 focused on the impact of thermal conductive enhancers (TCE) in the PCM on the solid-liquid phase change process and associated heat transfer. The set up was the same that was used in Chapter 2 including the heat source configuration and the imposed inlet conditions of the experiments. Four different nanoparticles (Silver, CuO, Al$_2$O$_3$ and Multi-walled carbon nanotubes) were used initially to find a comparatively best nanoparticle for
further study. Initial results show that in general, all TCE improved the heat transfer rate significantly in the PCM compared to net PCM. The magnitude of the heat flux increase relative to that for the plain wax varied approximately between 7-20%. Also the melting time reduced significantly and the results showed that almost 40% less time was required to melt half of the PCM volume. Although the rate declined to 25% for higher vertical height. CuO and silver nanoparticle showed better convective heat transfer coefficient increment than the other two. CuO nanoparticle among the four nanoparticle was considered for the rest of the study after experiencing some issues with the sinking/floatation and/or agglomeration of MWCNT and Al₂O₃ resulting in lower heat transfer in the PCM, whereas the issue of cost for silver nanoparticle. CuO at various mass fraction showed interesting results. Among the five different mass fractions of CuO were tried, CuO with 6% mass fraction showed better results. At closer distance from the heat source, the melting time of the PCM mixture with 6% CuO mass fraction showed melting rate 30% quicker than that for 1% CuO mass fraction. The heat transfer coefficient shows higher values at the lower fraction of the melting volume.

5.2 Contributions

In the present research, for the first time, a detailed investigation of the transient behaviour of the melting process and the dynamic of the solid-liquid interface was conducted.

First detailed investigation and analysis of the effects of heat source’s geometry, orientation and shape on the heat transfer performance and interface dynamics in a transient way. The outcome of this work has a direct impact on the design improvement of heat source/sink geometry in a PCM thermal storage, which has a direct impact on the overall thermal performance of the storage system.

Developed new correlation to characterize the convective heat transfer in the PCM during the melting process.

First detailed investigation of the influence of nanoparticles, acting as thermal conductive enhancers, on the phase change behavior and the associated heat transfer in a transient manner.
5.3 Recommendations for future work

1) 2D to 3D approach

The present study investigated the heat transfer process and the convective flow regime during the phase transition period in a PCM. The storage chamber, used during the study was constructed in such a fashion that the depth was kept very small compared to the other two dimensions. Hence, the chamber was assumed to have a two-dimensional geometry. This enabled to simulate the heat transfer and capture the interface dynamics during the melting process, which otherwise would have been extremely challenging to capture. In practical applications, all thermal storage systems have three-dimensional geometries and hence the solid-liquid interface as well as the heat transfer progresses in a three-dimensional manner. Therefore, the approach used in this study needs to be extended to three-dimensional storage chambers as well as three-dimensional geometries of the heat source/sink.

2) Solidification/freezing cycle

The present study was performed solely on the melting process of the PCM which enabled to grasp the knowledge of the heat storing process in the storage chamber. But a thermal storage operates in a cyclic manner i.e. the heat storage followed by a heat extraction. Hence, the investigations should be extended to include the investigation of the phase change process and associated heat transfer during the solidification/freezing process.

3) Pressure buildup and thermal expansion during melting

In the present research, paraffin wax was used as the PCM which has a volumetric expansion of approximately 10% when transformed from solid to liquid. This was a problem during the study when the heat source was near the bottom (chapter 2 and 4) since the melting was initiated from the bottom. The liquid formed in this region caused an expansion of the volume which in turn developed pressure inside the chamber. This
resulted liquid PCM leakage or squeezing the rubber gasket and probably the PCM bodily moved up. The problem was not that intricate since the experimental chamber was small. But this issue needs to be taken care in future works if the storage volume is bigger and a 3D chamber is used.

4) Nanoparticle suspension

In chapter 4, the study was extended to use nanoparticles with PCM to enhance the heat transfer rate. The main problem faced was the issues of suspension of the nanoparticles in the liquid PCM (nanofluid). Most conductive nanoparticles are heavier than the base fluid and as such this issue needs to be focused in future works.
Appendices

Appendix A

Appendix A: Methodology

An experimental approach is used to conduct research and meet thesis objectives. Detailed descriptions of the relevant experimental setup and measurement procedures are described in each Chapter. A brief description of general measurement techniques used in this research is provided below.

A.1 Experimental measurement techniques

Two different measurement techniques were employed simultaneously to gather data from the experimental runs, which were temperature measurements and image acquisitions. A set of T-type thermocouples (T30-2-507, Zesta Engineering Ltd.) were used to measure temperatures at various locations within the experimental apparatus that include the heat source and the PCM storage domain. The temperature data from all thermocouples were recorded simultaneously via a set of data acquisition cards (National Instrument’s thermocouple modules: NI-9214 and NI-9211) controlled by the LabVIEW software.

An imaging system was employed to take images and to quantify the melting process and the interface behaviour. A diffused lighting arrangement was made to provide a uniform and brighter illumination of PCM domain during imaging, which produced a good contrast between the solid and liquid domain, and also between the tracer particles inside the fluid and the background chamber wall. Hence, it helped to track the position of the interface by distinctly separating and identifying the solid and the liquid regions. A high resolution 4 Mega Pixel CCD camera (Flare 4M180-CL) with the resolution of 2048 × 2048 pixels was employed to capture the images. The camera was connected to an image acquisition system (DVR express core, IO Industries) to record images. The camera was controlled via the DVR express CoreView software installed on a computer.
A.2 Image processing

As this research involved the visualization and quantification of the melting process in a PCM, where it began with a single solid phase; transformed into the coexisting two-phase state (solid-liquid) and finally acquired a single liquid phase upon complete melting. Hence, the accurate tracking and quantification of the dynamic solid-liquid interface was a crucial part of this research.

The solid-liquid interface was detected in each image using in-house image processing and analysis algorithm developed in the Matlab environment. The computer code implemented image processing routines to detect the interface based on the change in gray-scale values between solid and liquid phases in each image. The main steps of the image-processing algorithm are shown in figure A.1. A sample original image of the PCM domain is shown in figure A.1(a). In the image, the white region represents the solid phase and dark gray region represents the melted wax. The improvement of the signal-to-noise ratio is the first step of the image-processing algorithm, which is achieved by rescaling the gray-values based on the maxima and minima. The image is segmented into a binary image in the next step by applying a threshold based on the gray-value distribution. Figure A.1(b) shows the corresponding image after segmentation. A series of morphological operations were performed on the binary image that include image inversion, dilation, filling the holes and erosion to remove noise from the binary image. Figure A.1(c) shows the final form of the processed image. The coordinates of the detected interface were recorded at every spatial location. The detected interface is plotted in figure A.1(d) over the original image for comparison. By processing a series of consecutive images from the beginning to the end of the melting process, a time history of the interface movement was obtained. This information was used in the further analysis of data. Note that the interface movement in the immediate vicinity of a thermocouple was locally disturbed. Hence, the interface data in the vicinity of each thermocouple were excluded from subsequent analyses.
Figure A.1: Sequence of the processes involved to generate the solid-liquid interface of the PCM; (a) original image of the PCM domain (thermocouples are seen by white circles), (b) image after adjustment and converting into the binary image, (c) image after inversion, dilation, filling the holes and erosion and (d) image with the instantaneous interface shown as the red line
Figure B.1: Schematics of the storage chamber showing the locations of the insulation
Appendix C

Appendix C: Experimental Uncertainty Analysis

All experimental measurement systems involve some errors which is actually the difference between the true value of the variable and the value assigned by the measurements. But, in any measurement, in fact, the true value is unknown. Thus, instead of the actual error, we estimate the probable error in the measurement which is termed as the uncertainty. Hence, the uncertainty defines an interval about the measured variable within which the assumed true value must fall.

Any kind of experimental measurements involve some level of experimental uncertainty. These uncertainties may originate from various causes such as inaccuracy in measurement equipment, random variations in the quantities measured and the approximations considered in the data (data reduction). It is impossible to eliminate the errors completely and hence it is very important to know the level of uncertainty for a given measurement. Since in a measurement system the errors may be contributed from various sources, they all need to be considered. Thus, all these uncertainties in individual measurements are eventually accommodated into the uncertainty in the final results.

The errors are of two general categories, (1) Systematic or bias errors and (2) Precision or random errors.

1. Systematic or bias errors

   Systematic or bias errors are caused from a drift or shift of sample mean from the true mean by a fixed amount.

2. Precision or random errors.

   This error is caused by a lack of repeatability in the output of a measurement system. The random errors bring a distribution of measured values about the sample mean.

General equation used for experimental uncertainty calculation for precision errors:

\[ W_R = \left[ \sum_{i=1}^{n} \left( W_{x_i} \frac{\partial R}{\partial x_i} \right)^2 \right]^{1/2} \]
Data acquisition errors: sources

a) T-type thermocouples (T30-2-507, Zesta Engineering Ltd.): uncertainty ±0.4%.

b) Air flow meter, Omega FL-2000 series: accuracy as ±5%.

c) PID controller (Zesta Engineering Ltd. ZEL-9100): accuracy ±2°C.

d) HTF tube dimensions: digital sliding calliper (Mastercraft, model 58-6800-4)

Sample uncertainty calculation for heat flux, \( q'' \): HTF temperature 175 °C:

Following equation is used to compute the heat flux:

\[
q'' = \frac{\dot{m} C_p (T_{Surf} - T_{Bulk})}{\pi D l} = \frac{\rho V C_p (\Delta T)}{A}
\]

(C1)

Where,

\( \rho \) = Avg. density of HTF: 0.868 ±2.28% = 0.868±0.0198 kg/m³

\( V \) = Volume flow rate of HTF: 0.00067±5% = 0.00067±0.0000335 m³/s

\( C_p \) = Specific heat of HTF: 1013±2.28% = 1013±23.09 J/kg.K

\( \Delta T \) = Heat source and bulk PCM temp. diff.: 26.86±0.533 °C

\( T_{Setting} \) = Temp. setting in controller: 175±2 °C

\( A \) = Heat source surface area: 0.001496±0.00000314 m²

From equation (1), finding the value of \( q'' \),

\( q'' = 0.868 \times 0.00067 \times 1013 \times 26.86 / 0.001496 = 10577.38 \text{ W/m}² \)

General equation to estimate the uncertainty:

\[
W_{q''} = \left[ \sum_{i=1}^{n} \left( W_{x_i} \frac{\partial q''}{\partial x_i} \right)^2 \right]^{1/2}
\]

(C2)

Applying general equation to estimate the individual uncertainty:

\[
W_{q''} = \left[ \left( W_{\rho} \frac{\partial q''}{\partial \rho} \right)^2 + \left( W_{V} \frac{\partial q''}{\partial V} \right)^2 + \left( W_{C_p} \frac{\partial q''}{\partial C_p} \right)^2 + \left( W_{\Delta T} \frac{\partial q''}{\partial \Delta T} \right)^2 + \left( W_{T_{set}} \frac{\partial q''}{\partial T_{set}} \right)^2 + \left( W_{A} \frac{\partial q''}{\partial A} \right)^2 \right]^{1/2}
\]
Now computing each term separately,

\[
\frac{\partial q^*}{\partial \rho} = \frac{V C_p (\Delta T)}{A} = \frac{0.00067 \times 1013 \times 26.86}{0.001496} = 12815.92
\]

\[
\left( W_\rho \frac{\partial q^*}{\partial \rho} \right) = 0.0198 \times 12815.92 = 241.28
\]

\[
\frac{\partial q^*}{\partial V} = \frac{\rho C_p (\Delta T)}{A} = \frac{0.868 \times 1013 \times 26.86}{0.001496} = 15787144.55
\]

\[
\left( W_V \frac{\partial q^*}{\partial V} \right) = 0.0000335 \times 15787144.55 = 528.87
\]

\[
\frac{\partial q^*}{\partial C_p} = \frac{\rho V (\Delta T)}{A} = \frac{0.868 \times 0.00067 \times 1013}{0.001496} = 10.44
\]

\[
\left( W_{C_p} \frac{\partial q^*}{\partial C_p} \right) = 23.09 \times 10.44 = 241.09
\]

\[
\frac{\partial q^*}{\partial \Delta T} = \frac{\rho V C_p}{A} = \frac{0.868 \times 0.00067 \times 1013}{0.001496} = 393.79
\]

\[
\left( W_{\Delta T} \frac{\partial q^*}{\partial \Delta T} \right) = 0.533 \times 393.79 = 209.89
\]

\[
\frac{\partial q^*}{\partial T_{set}} = \frac{\rho V C_p}{A} = \frac{0.868 \times 0.00067 \times 1013}{0.001496} = 393.79
\]

\[
\left( W_{T_{set}} \frac{\partial q^*}{\partial T_{set}} \right) = 2 \times 393.79 = 787.59
\]

\[
\frac{\partial q^*}{\partial A} = -\frac{V C_p (\Delta T)}{A^2} = -\frac{0.868 \times 0.00067 \times 1013 \times 26.86}{(0.001496)^2} = -7070445.75
\]

\[
\left( W_A \frac{\partial q^*}{\partial A} \right) = 0.00000314 \times 7070445.75 = 22.20
\]

Now,

\[
W_{q^*} = [241.28^2 + 528.87^2 + 241.09^2 + 209.89^2 + 787.59^2 + 22.20^2]^{1/2}
\]

\[
W_{q^*} = 1029.99 \text{ W/m}^2
\]

Therefore, \( q^* = 10577.38 \pm 1029.99 \text{ W/m}^2 \)

Or, \( q^* = 10577.38 \pm 9.73\% \text{ W/m}^2 \)

**Reference**

Appendix D

Figure D.1: Circuit diagram for temperature acquisition via Labview program
Curriculum Vitae

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Post-secondary Education and Degrees:

Bangladesh University of Engineering and Technology (BUET) Dhaka, Bangladesh
B.Sc. Engineering (Mechanical) 1981-1985

The University of Western Ontario (UWO) London, Ontario, Canada
Master of Engineering, Mechanical and Materials Engineering 2008-2010

The University of Western Ontario (UWO) London, Ontario, Canada
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Honors and Awards:

Queen Elizabeth II Graduate Scholarship in Science and Technology (QEII-GSST) 2011-2012

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Related Work Experience:

Teaching Assistant The University of Western Ontario 2011-2016

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ASME: 2016 Summer Heat Transfer Conference (HT/FE/ICNMM), July 2016, Washington DC, USA
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